The Association
of
Engineering and Shipbuilding
Draughtsmen.

GEAR MANUFACTURE.

BOOK 5.

Worm Gear Detail Design.
Worm Gear Manufacture.
Hob Design for Spur, Spiral and Helical Gears.
Gear Shaper Cutter Design for Worms and Screws.

By F. WHITELEY.

Published by The Association of Engineering and Shipbuilding Draughtsmen,
96 St. George's Square, London, S.W.1.

SESSION 1951-52.

Printed by Milne, Tannahill & Methven, Ltd. (T.U.), 12-14 Mill Street, Perth.

ADVICE TO INTENDING AUTHORS OF A.E.S.D. PRINTED PAMPHLETS.

Pamphlets submitted to the National Technical Sub-Committee for consideration with a view to publication in this series should not exceed 9,000 to 10,000 words and about 20 illustrations, making a pamphlet of about 40 to 48 pages. The aim should be the presentation of the subject clearly and concisely, avoiding digressions and redundance. Manuscripts are to be written in the third person.

Drawings for illustrations should be done either on a good plain white paper or tracing cloth, deep black Indian ink being used. For ordinary purposes they should be made about one-and-a-half times the intended finished size, and it should be arranged that wherever possible these shall not be greater than a single full page of the pamphlet, as folded pages are objectionable, although, upon occasion, unavoidable. Where drawings are made larger, involving a greater reduction, the lines should be made slightly heavier and the printing rather larger than normal, as the greater reduction tends to make the lines appear faint and the printing excessively small in the reproduction. In the case of charts or curves set out on squared paper, either all the squares should be inked in, or the chart or curve should be retraced and the requisite squares inked in. Figures should be as self-evident as possible. Data should be presented in graphical form. Extensive tabular matter, if unavoidable, should be made into appendices.

Authors of pamphlets are requested to adhere to the standard symbols of the British Standards Institution, where lists of such standard symbols have been issued, as in the case of the electrical and other industries, and also to the *British Standard Engineering Symbols and Abbreviations*, No. 560, published by the B.S.I. in 1934 at 5/-. Attention might also be given to mathematical notation, where alternative methods exist, to ensure the minimum trouble in setting up by the printer.

The value of the pamphlet will be enhanced by stating where further information on the subject can be obtained. This should be given in the form of footnotes or a bibliography, including the name and initials of the author, title, publisher, and year of publication. When periodicals are referred to, volume and page also should be given. References should be checked carefully.

Manuscripts, in the first instance, should be submitted to the Editor, *The Draughtsman*, 96 St. George's Square, London, S.W.1.

For Pamphlets, a grant of £20 will be made to the Author, but special consideration will be given in the case of much larger pamphlets which may involve more than the usual amount of preparation.

The Publishers accept no responsibility for the formulae or opinions expressed in their Technical Publications.

The Association
of
Engineering and Shipbuilding Draughtsmen.

GEAR MANUFACTURE.

BOOK 5.

Worm Gear Detail Design.
Worm Gear Manufacture.
Hob Design for Spur, Spiral and Helical Gears.
Gear Shaper Cutter Design for Worms and Screws.

By F. WHITELEY.

Published by The Association of Engineering and Shipbuilding Draughtsmen,
96 St. George's Square, London, S.W.1.

SESSION 1951-52.

4/52.

BOOK 5.

CONTENTS.

Page

Chapter 1. Worm Gear Detail Design - - - - **5-26**

Chapter 2. Worm Gear Manufacture - - - - **27-33**

Chapter 3. Hob Design for Spur, Spiral and Helical Gears - - - - - - **33-44**

Chapter 4. Gear Shaper Cutter Design for Worms and Screws - - - - **45-51**

GEAR MANUFACTURE.

By F. WHITELEY.

CHAPTER I.

WORM GEAR.

DETAIL DESIGN.

IN the design of worm gearing two entirely different types have to be considered, viz. :—

(1) Ordinary worm gearing
(2) Module system of worm gearing

the first being generally used for applications, in which the principal requirement is a high ratio of reduction, and where efficiency and load carrying capacity are not the most important considerations, as, for example, feed drives in machine tools.

The second type is used when greater efficiency is required for high speed drives, and where the load carrying capacity is of primary importance, as is the case for automobile rear axle drives, and spindle drives for machine tools, etc.

Both types, however, embody a parallel type of worm and constitute a direct development of the simple screw, the only difference between the two types being found in the form of thread, and thread proportions.

Ordinary worm gearing is based upon the analogy between a screw worm and wheel, and a spur rack and pinion, and forms a convenient basis for fixing the tooth proportions, as shown in Fig. 1.

The worm is made straight sided on the axial section, the pressure angle on this section being dependent on the number of teeth in the wheel, and to prevent undercutting of the wheel flanks, the minimum number of teeth

$$T = \frac{2}{\sin^2 \psi_a}$$

when T = minimum number of teeth in worm wheel.
ψ_a = axial pressure angle of worm.

As a general guide, however, and for worm gears of standard proportions with helix angles of not more than 20° the design should satisfy the following conditions :—

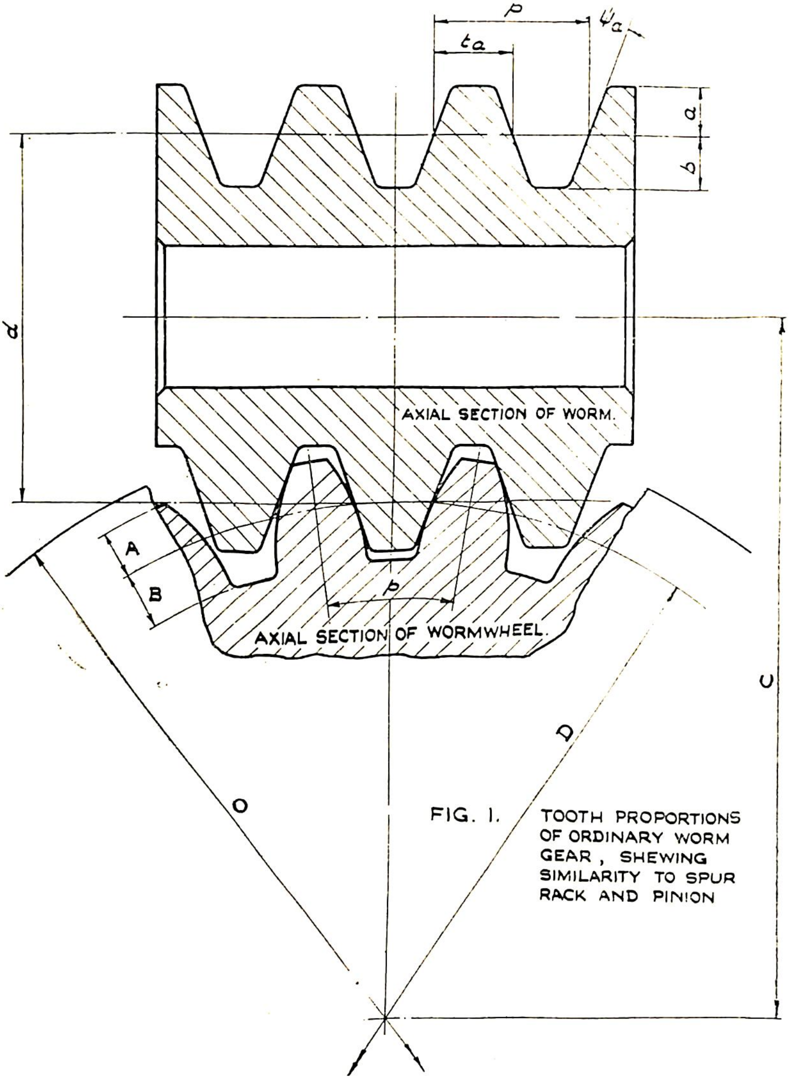


FIG. 1. TOOTH PROPORTIONS OF ORDINARY WORM GEAR, SHEWING SIMILARITY TO SPUR RACK AND PINION

32 teeth and over $\psi_a = 14\frac{1}{2}°$ pressure angle.
30 ,, ,, ,, $\psi_a = 15°$,, ,,
17 ,, ,, ,, $\psi_a = 20°$,, ,,
11 ,, ,, ,, $\psi_a = 25°$,, ,,

The tooth proportions for this type of worm gearing are based on the circular pitch standard, due to the fact that in the past the majority of worms were produced on a centre lathe in the same manner as used for a screw, and given below is the notation and formula for gears running at standard centre distances.

When the helix angle is greater than 20°, however, it is advisable to revert to the module system for the detail design, although some designers favour the practice of making the addendum and dedendum of both worm and wheel in terms of the normal pitch, and the normal pressure angle in accordance with the angles specified above. This latter practice is not to be recommended, however, as it is likely to create difficulties in manufacture.

p = axial pitch of worm and circular pitch of worm wheel.
a = addendum of worm.
A = throat addendum of wheel.
b = dedendum of worm.
B = dedendum of wheel.
d = pitch circle diameter of worm.
D = pitch circle diameter of wheel.
j = outside diameter of worm.
J = throat diameter of wheel.
O = overall diameter of wheel.
C = centre distance.
G = throat or gorge radius of wheel.
F_p = face width of worm.
F_w = face width of wheel.
θ = helix angle of worm thread.
p_n = normal pitch of worm.
L = lead of worm.
t_n = normal thickness of worm thread.
t_a = axial thickness of worm thread.
ψ_a = axial pressure angle of worm thread.
ψ_n = normal pressure angle of worm thread.
T = number of teeth in wheel.
t = number of threads or starts in worm.
R = ratio.
a_c = chordal addendum of worm thread normal plane.
t_c = chordal thickness of worm thread normal plane.

Dimension Required	*Particulars Given*	*Formula*
d (approx.)	C, R and θ	$\dfrac{2C}{1+R\tan\theta}$
D (nominal)*	C and d	$2C - d$
D (actual)	p and T	pT/π
p	D and T	$\pi D/T$
C	D and d	$\dfrac{D+d}{2}$
L	p and t	pt
tan θ	d and L	$L/\pi d$
p_n	p and θ	$p\cos\theta$
a or A	p	$0{\cdot}3183p$
b or B	p	$0{\cdot}3683p$
a	p_n } if θ exceeds 20°	$0{\cdot}3183p_n$
b	p_n	$0{\cdot}3683p_n$
j	d and a	$d + 2a$
J	D and A	$D + 2A$
O	D and A	$D + 3A$
F_p	p	$4p$
F_w (Max)	d and a	$2\sqrt{a^2+da}$
R	T and t	T/t

* D (nominal) and D (actual) are identical when centres are standard, if centres are increased D (nominal) must be used for all calculations.

Another method which can be used to eliminate undercutting of the wheel flanks is to increase the centres slightly by an amount "e" as shown in Fig. 2, but care should be taken to ensure that the increase is not excessive, or the crests of the wheel teeth may become too pointed.

The position of the worm pitch line is determined only by the position of the pitch circle diameter of the wheel. If, therefore, the centre distance of the gears is increased keeping all the dimensions of the worm the same, the position of the worm pitch line is brought nearer to the crest of the worm threads, and the worm addendum thereby reduced.

In making any calculations, however, in connection with increased centre distance gearing it will be found convenient to base the detail dimensions of the wormwheel on the nominal pitch circle diameter, which, as previously stated for a worm wheel meshing at a standard centre distance is equal to

$$D = 2C - d$$

whereas for extended centre distance worm gearing the actual pitch circle diameter is equal to

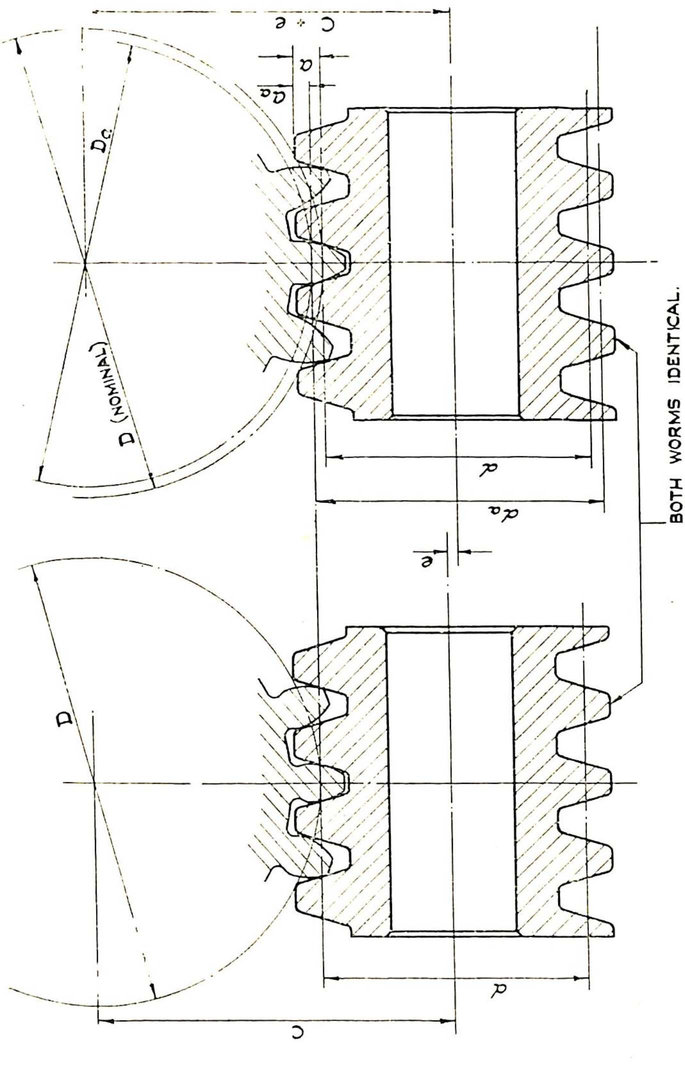


FIG. 2. SHEWING EFFECT OF INCREASE IN CENTRE DISTANCE ON WHEEL TOOTH SHAPE.

$$D_a = Tp/\pi.$$

and of the worm $d_a = 2C - D_a$

The actual addendum of the worm measured from the actual pitch diameter d_a should not exceed

$$a_a = \frac{D_a \sin^2 \psi_a}{2}$$

therefore the distance must be extended to avoid undercutting by an amount equal to

$$e = a - \frac{D_a \sin^2 \psi_a}{2}$$

when $a = 0{\cdot}3183p$

from which $C = \dfrac{d + D_a}{2} + e$

thus the nominal pitch circle diameter of the wheel from which the blank dimensions are calculated will be

$$D = D_a + 2e$$

In well balanced design of worm gearing, the gears should have an equal margin of safety to withstand the breaking stress on the worm wheel teeth, the surface pressure on the teeth, and the bending stress on the wormshaft.

At the same time the wormshaft should be sufficiently stiff to avoid serious deflection, but if made larger than is necessary, this will lead to increased sliding velocity and reduced efficiency.

The number of teeth in the wormwheel again depends upon the conditions of loading, and in many industrial drives, for example, the rating of worm gears is determined not by the strength of the gears themselves, but by the ability of the housing to radiate the heat generated by friction.

The pitch will then be smaller and the general design lighter than is the case for gears of similar centre distance, and ratio, which, through auxiliary cooling or intermittent loading, may be called upon to transmit the maximum possible power.

Bearing in mind these various factors the following tables may be used for industrial drives only, as a guide in selecting the pitch diameter of the worm, and the number of teeth in the wormwheel.

C	*d*	T	C	*d*	T	C	*d*	T
3	1·5	26	9	3·0	43	30	6·4	71
4	1·8	30	10	3·2	45	35	7·0	76
5	2·1	33	12	3·6	48	40	7·8	80
6	2·3	36	15	4·1	53	60	10·0	95
7	2·5	38	20	5·0	60			
8	2·8	40	25	5·7	66			

The number of threads in the worm is found by dividing the approximate number of teeth in the wheel T by the ratio R required.

There will usually be various alternatives some giving a larger, and others a smaller number of teeth in the wheel than that assumed for the preliminary design.

The final choice will, however, depend upon whether the gears are likely to be heavily stressed, and also bearing in mind that there should be no common factors between the number of threads in the worm and the number of teeth in the wheel.

Example.

To find the centre distance and wheel blank dimensions for a single start worm 2·500 inches pitch diameter 0·75 inches axial pitch, 15° axial pressure angle, 0·239 addendum, 0·276 dedendum, and meshing with an 18 tooth wormwheel, we proceed as follows :—

Actual wheel pitch diameter.

$$D_a = \frac{Tp}{\pi} = \frac{18 \times 0{\cdot}75}{3{\cdot}1416} = 4{\cdot}297$$

To avoid undercutting of wheel tooth flanks, actual worm addendum must not exceed

$$a_a = \frac{D_a \sin^2 \psi_a}{2} = \frac{4{\cdot}297 \sin^2 15^\circ}{2} = 0{\cdot}144$$

Increase in centre distance "e"

$$= a - a_a = 0{\cdot}239 - 0{\cdot}144 = 0{\cdot}095$$

$$\text{and } C = \frac{d + D_a}{2} + e = \frac{2{\cdot}500 + 4{\cdot}297}{2} + 0{\cdot}095$$

$$= 3{\cdot}493$$

$$D = D_a + 2e$$

$$= 4{\cdot}297 + 2 \times 0{\cdot}095 = 4{\cdot}487$$

$$\text{and } J = D + 2A = 4{\cdot}487 + 2 \times 0{\cdot}239 = 4{\cdot}965$$

$$O = D + 3A = 4{\cdot}487 + 3 \times 0{\cdot}239 = 5{\cdot}204$$

$$F_w = 2\sqrt{a^2 + da} = 2\sqrt{0{\cdot}239^2 + 2{\cdot}500 \times 0{\cdot}239}$$

$$= 1{\cdot}618 \text{ say } 1{\cdot}625$$

and for the worm

$$j = d + 2a = 2{\cdot}500 + 2 \times 0{\cdot}239 = 2{\cdot}978$$

$$F_p = 4_p = 4 \times 0{\cdot}75 = 3{\cdot}000$$

Vernier Settings.

Vernier settings are usually only calculated for the worm, a suitable hob or fly hob being made to cut the wheel, this hob being set at correct cutting centres depending upon the hob diameter, after which the wheel is checked with the worm at correct running centres for backlash, etc.

Then for worm

a_c = chordal addendum
t_c = chordal thickness

and $a_c = a$ (for helix angles up to 20°)

$$\text{when } \tan\theta = L/\pi d = \frac{0{\cdot}750}{3{\cdot}1416 \times 2{\cdot}500} = 5^\circ - 27'$$

$$\text{then } a_c = 0{\cdot}239$$

$$t_c = pn/2 - \text{total backlash}$$

$$\text{when } p_n = p\cos\theta = 0{\cdot}75 \times 0{\cdot}99548$$

$$= 0{\cdot}7466$$

$$t_c = \frac{0{\cdot}7466}{2} - \text{total backlash}$$

$$= 0{\cdot}343 - \text{total backlash}$$

The normal tooth thickness of the hob used to cut the wheel should then be made equal to $p_n/2$ as all the backlash has been deducted from the worm thickness.

The total backlash allowed should be the same as that specified for spur gears in Book 2, Chapter 1.

The gorge or throat radius required on the centre of the wheel face is produced by the hob used for cutting the wheel teeth, but if a fly hob is used for cutting it is then necessary to machine this radius on the lathe, and in this case G must be calculated.

$$G = C - J/2 \text{ and in example}$$

$$G = 3{\cdot}493 - \frac{4{\cdot}965}{2} = 1{\cdot}011$$

Module System.

The second type of worm gearing to be considered is the module system, and this differs from the first type only as far as the form of thread and tooth proportions are concerned.

A worm of this type can, in fact, be likened to a spiral gear produced by the generating method as both are of involute tooth form on the transverse section, due to the fact that the tooth surfaces of both are produced from a normal basic rack.

Where, however, the tooth proportions of a spiral gear are made standard in terms of the normal diametral pitch, the module system worm tooth proportions are based primarily on the module of the worm in inches, but slight variations may be required depending upon the number of teeth in the worm wheel.

The module system is used, however, in an endeavour to limit the number of worms (and thereby the number of hobs used for cutting the wormwheels) which may ultimately be made, and in addition to having certain advantages over the circular pitch system, is also free from recurring decimals, usually met with when calculating the pitch diameters associated with many diametral pitches.

The module, in the case of a worm wheel is simply the pitch diameter divided by the number of teeth, and conversely the pitch diameter is equal to the number of teeth multiplied by the module.

This system has been adopted by the British Standards Institution in their specification No. 721—1937, and offers a convenient means of standardisation, as the actual pitch diameter of the wheel and the nominal pitch diameter of the worm are made exact multiples of the module.

To obtain the worm pitch diameter a diameter quotient denoted by the symbol q is introduced from which worms are standardised, each being completely defined by :—

(*a*) the number of threads or starts.
(*b*) the module in inches (chosen from a standard range).
(*c*) the number of modules in the worm pitch diameter (termed the diameter quotient q).

All worms, therefore, having the same number of starts and the same value of q are then geometrically similar, and their respective dimensions are directly proportional to the module selected.

In other words the nominal worm pitch diameter is simply equal to q times the module, *i.e.*, the nominal worm pitch diameter is equal to that of a spur gear or worm wheel of the same module pitch and having q teeth.

By this method the lead angle of the worm is also standardised, but in a manner which completely eliminates angular measurements (apart from the pressure angle) from the calculations.

The values of q which are best adapted to ordinary practice are found to range for purposes of standardisation from 6 to 10 inclusive, values below and above this range being used only in special cases.

For each pitch or module and number of threads it is possible to have five dimensionally different worm forms, thus for worms of one to five threads inclusive the total number of different worms would be equal to 25 for each pitch per hand.

Should the range be extended to include worms up to 10 threads the total number of worms it would be possible to obtain is 40, since q should be less in value than t, but when selecting the value for q it should be higher for a slow speed heavily loaded gear than for one running at a high speed and only lightly loaded.

In order, however, to take the fullest advantage of the possibility of reducing to a minimum the number of hobs by using a series of standard worm forms the number of modules should also be reduced to a minimum.

In some designs, the centre distance, numbers of teeth, and pitch diameters are so closely restricted, especially in cases where an existing drive is being replaced by a different ratio, that a module pitch which is a multiple of 0·1 or even 0·05 inches cannot be used.

Owing to the flexibility of centres of a worm gear, however, a standard integral value of q can almost invariably be worked in, together with a module pitch which is multiple of 0·025 inches.

In extreme cases, however, it is more important to use a whole number for q than use a standard module, and a list of modules which can be regarded as standard is given below :—

from 0·1 to 0·2 in increments of 0·025
,, 0·2 to 0·5 ,, ,, ,, 0·05
,, 0·5 to 0·8 ,, ,, ,, 0·10
,, 1·0 to 2·0 ,, ,, ,, 0·25

Whatever value of q is selected, whole number or otherwise, and given the centre distance, the module may then be found from

$$M = \frac{2\,C}{T+q} \quad \text{when}$$

M = module in inches
C = centre distance
T = number of teeth in wormwheel
q = diameter quotient

and a standard value of M can usually be brought in by the adjustment of either the centre distance or the value of q.

If, however, values for T, q and M are given or assumed the centre distance can then be found from

$$C = \frac{M\,(T+q)}{2}$$

which will be recognised as the same formula as was used for calculating the centre distance of a pair of spur gears using the module system, q being substituted for the number of teeth in the pinion.

To define the proportions of the worm, the designation $t/q/M$ is used and this value should be stamped or etched on the worm when

t = number of threads or starts in worm
q = diameter quotient
M = module

It is also advisable to stamp similar particulars on the worm wheel, in this case the number of teeth being added, viz. :—

$$\frac{t}{T}\,/\,q\,/\,M$$

when T = number of teeth in the wormwheel.

To restrict the number of worms still further it is recommended that right-hand worms should be used wherever possible, and thus increase the utility of any hobs previously made in accordance with the system.

Tooth Proportions.

A constant worm addendum is used throughout and this enables a quick check to be made on the outside diameter of the worm, and also facilitates the calculation of q when the overall diameter is limited.

The worm addendum is made equal to the module then

$$a = \mathrm{M}$$

and as the whole depth of tooth varies according to the lead angle, this produces varying proportions of addendum to dedendum, for as the lead angle increases for a given module, the dedendum of the worm diminishes, but the addendum remains the same.

then $b = 2{\cdot}2\ \mathrm{M_n} - a$
when $\mathrm{M_n} = \mathrm{M} \cos \theta$
and M = module
θ = lead or helix angle
a = worm addendum
b = worm dedendum
or $a + b = 2{\cdot}2\ \mathrm{M_n}$

The normal tooth thickness is also based on the normal module but as the axial tooth thickness is made one-half the axial pitch at the mean depth (including clearance).

$$\tan \theta_m = \frac{\mathrm{L}}{d_m \pi} \text{ when } d_m = i + \frac{(j-i)}{2} \text{ and } t_n = \frac{\pi \mathrm{M}}{2} \cos \theta_m$$

when L = lead
M = module
θ_m = helix angle at mean depth
t_n = normal tooth thickness at mean depth
j = outside diameter
i = root diameter
d_m = pitch diameter at mean depth

Pressure Angle.

It has been previously stated that the tooth surfaces of the worm thread are generated in the same way as is used for a spiral gear, and in the system a normal basic rack having a pressure angle of 20° is used as standard.

Before, however, this pressure angle can be definitely established the design should satisfy the following conditions :—

1. The base diameter should not exceed the root diameter plus twice the clearance, and if this is the case it is then necessary to increase the basic normal pressure angle accordingly.

If therefore the base diameter exceeds the root diameter plus twice the clearance the base diameter is made equal to

$$d_o = i + 2_c$$

and $\cos \psi_t = \dfrac{d_o}{d}$

the new value for the basic normal pressure angle

$\tan \psi_n = \tan \psi_t \sin \theta$

when d_o = base diameter of worm

i = root diameter of worm

c = clearance

ψ_t = transverse pressure angle of worm

d = pitch diameter of worm

ψ_n = basic normal pressure angle of worm

θ = helix angle of worm

2. For worms having large helix angles meshing with wormwheels having small numbers of teeth, it may be necessary to increase both the root and base diameters in order to avoid too small a crest and root space widths, or the base and root diameters must be greater than the standard value given by $i = d - 2b$, with the base diameter $d_o = 2c + i$.

The sum of the worm crest and root space widths on the axial section should not be made less than 1·25M, therefore to satisfy this condition, and referring to Fig. 3,

$$\tfrac{1}{2}\,(\pi - 1{\cdot}25)\text{ M should be greater than } \frac{\text{L inv. }\psi_1}{2\,\pi}$$

when M = module

L = lead of worm

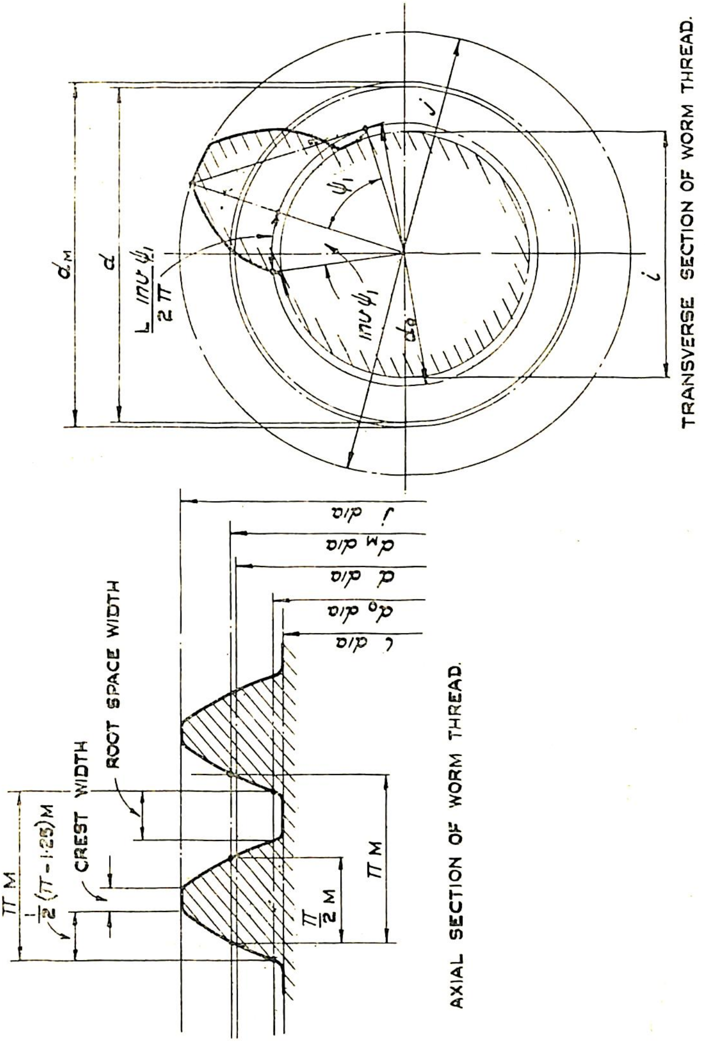
ψ_1 = transverse pressure angle of worm on outside diameter.

j = outside diameter of worm

d_o = base diameter of worm

inv. ψ_1, being an involute function of the angle ψ_1, which can be obtained from tables.

If the value $\frac{1}{2}\,(\pi - 1{\cdot}25)$ M is less than $\dfrac{\text{L inv. }\psi_1}{2\,\pi}$ the base and root diameters should be increased to obtain a new value for ψ_1,



TRANSVERSE SECTION OF WORM THREAD.

AXIAL SECTION OF WORM THREAD.

FIG. 3. DIAGRAM OF WORM THREAD INDICATING WORM CREST AND ROOT SPACE WIDTHS.

when $\cos \psi_1 = d_o/j$
and $\cos \psi_t = d_o/d$
from which $\tan \psi_n = \tan \psi_t \sin \theta$

when ψ_t = transverse pressure angle
ψ_n = basic normal pressure angle
θ = helix or lead angle of worm
d = pitch diameter of worm

Blank Dimensions of Worm.

For the detail design of standard worms the following formula is used :—

$\tan \theta = t/q$
$a = M$
$b = M\,(2{\cdot}2 \cos \theta - 1)$
$c = 0{\cdot}2\,M \cos \theta$
$d = q\,M$
$j = d + 2a$
$\quad = M\,(q + 2)$

$i = d - 2b$ (Check worm crest and root widths)
$\psi_n = 20°$
$\tan \psi_a = \tan \psi_n \sec \theta$

$$\tan \psi_t = \frac{\tan \psi_n}{\sin \theta}$$

$d_o = d \cos \psi_t$
(*Note*—Value should not exceed $i + 2c$)
$\cos \theta_o = \cos \theta \cos \psi_n$
$F\,p = 8\,M$
Note—Should the base diameter be altered new values will be required for ψ_n and ψ_t.

Blank Dimensions of Wormwheel.

$A = M\,(2 \cos \theta - 1)$
$B = M\,(0{\cdot}2 \cos \theta + 1)$
$D = 2C - q\,M$
$J = D + 2\,A$
$I = D - 2\,B$

$O = D + 2{\cdot}5\,A$
$F = 2\,M \sqrt{q + 1}$
$G = C - J/2$

when A = wheel addendum
B = wheel dedendum
D = nominal pitch diameter of wheel
J = throat diameter of wheel
I = root diameter of wheel
O = overall diameter of wheel
F_w = face width of wheel
G = throat or gorge radius

The terms used for both worm and wheel are indicated in Fig. 4.

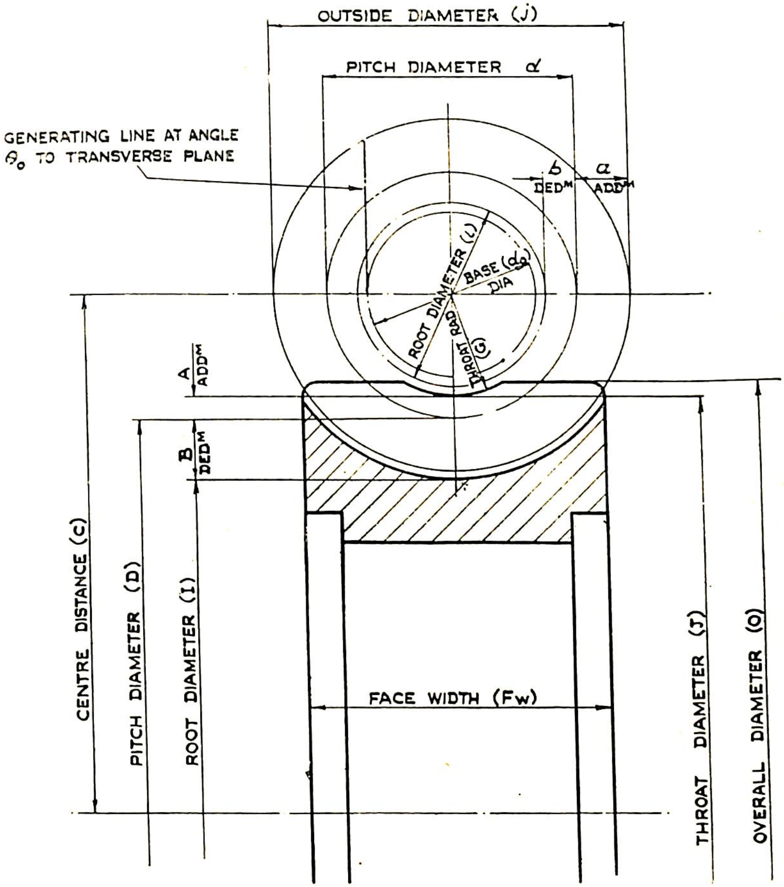


FIG. 4. DIAGRAM OF MODULE SYSTEM WORM AND WHEEL

Centre Distance Change.

When calculating blank dimensions of both worm and wheel the calculations must be based on the *nominal* pitch diameters, the nominal pitch diameter of the worm being equal to

$$d = q\,\mathrm{M}$$

and the nominal pitch diameter of the wheel

$$\mathrm{D} = \mathrm{TM}$$

the centre distance then being obtained from the formula

$$C = \frac{(T + q)\ M}{2}$$

Should, however, it be found necessary to make a slight change in the centre distance in order to use standard values for q or M the actual or working pitch diameters will then differ from the nominal pitch diameters.

In other words if no change is made in the centre distance from standard the nominal and actual pitch diameters will agree.

A good rule to remember when finding nominal pitch diameters is to work from the worm, and to find actual pitch diameters work from the wheel, *i.e.*,

$$\begin{aligned} d \text{ (nominal)} &= q\,M \\ D \text{ (nominal)} &= 2C - d \text{ (nominal)} \\ D \text{ (actual)} &= T\,M \\ d \text{ (actual)} &= 2C - D \text{ (actual)} \end{aligned}$$

The amount by which the centre distance can be departed from depends largely upon the number of teeth in the wheel, and the number of threads in the worm.

For example if t is small and T is less than 16 then undercutting may occur, but the number of teeth may be less than 16 with a larger number of threads, and a subsequent increase of lead angle.

On the other hand an increase in centre distance when T is small may produce a narrow crest width on the wheel teeth at the point where the throat radius meets the overall diameter of the wormwheel, but if T exceeds 22 then a change in centre distance of $\pm$ 0·25M (preferably plus) is usually permissible.

Vernier Settings for Worm.

Vernier settings are given for the worm only, the wheel being cut with a hob or fly hob corresponding to the worm at a specified centre distance.

The worm tooth is measured with a pair of gear tooth verniers on the normal plane, but as the axial thread thickness is made equal to half the axial pitch at the mean full depth of tooth it is necessary to calculate the height and thickness setting on the normal plane at the mean full depth of tooth.

Referring to Fig. 5 we have :—

$$d_m = i + \frac{(j - i)}{2}$$

$$\tan \theta_m = \frac{tM}{d_m} \quad \text{or} \quad \frac{L}{d_m\,\pi}$$

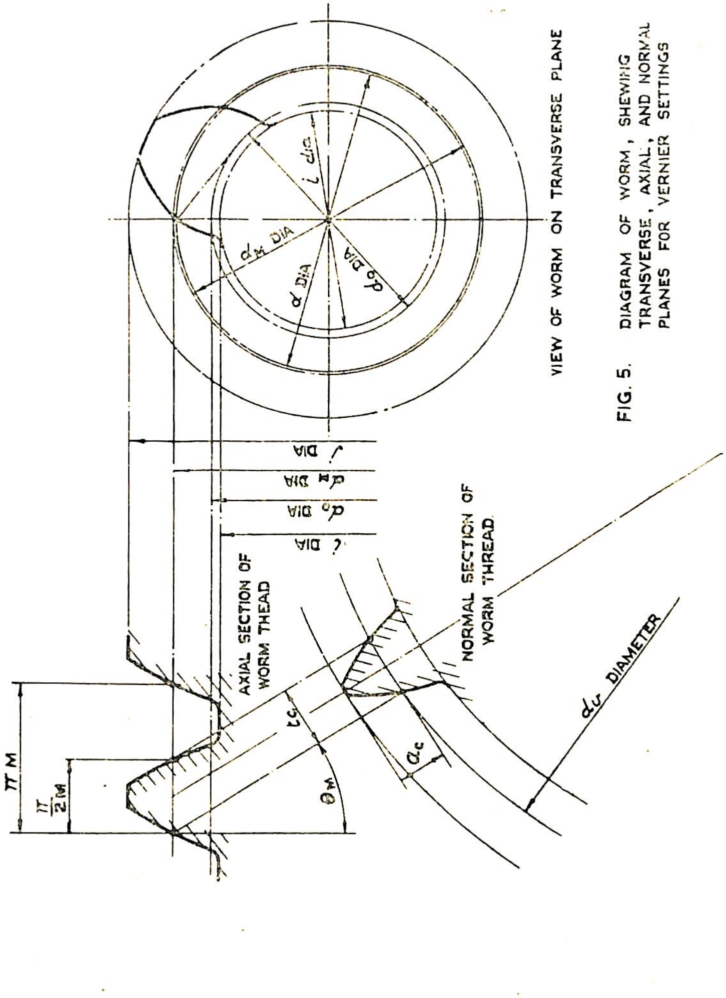


FIG. 5. DIAGRAM OF WORM, SHEWING TRANSVERSE, AXIAL, AND NORMAL PLANES FOR VERNIER SETTINGS

when d_m = diameter on mean full depth of tooth
θ_m = helix or lead angle at mean full depth of tooth
i = root diameter of worm
j = outside diameter of worm
t = number of threads or starts
M = module
L = lead of worm

then $t_c = \dfrac{\pi M}{2} \cos \theta_m$ — total backlash

and $a_c = \dfrac{j - d_m}{2}$

when t_c = chordal thickness of worm tooth on normal plane
a_c = chordal addendum of worm tooth

Theoretically the above values should be corrected slightly due to the fact that the points of measurement are actually on an arc, the diameter of which is equal to :—

$d_v = \dfrac{d_m}{\sin^2 \theta_m}$ when d_m = pitch diameter of worm at mean depth.
θ_m = helix angle of worm at mean depth.
d_v = virtual pitch diameter of worm at mean depth.

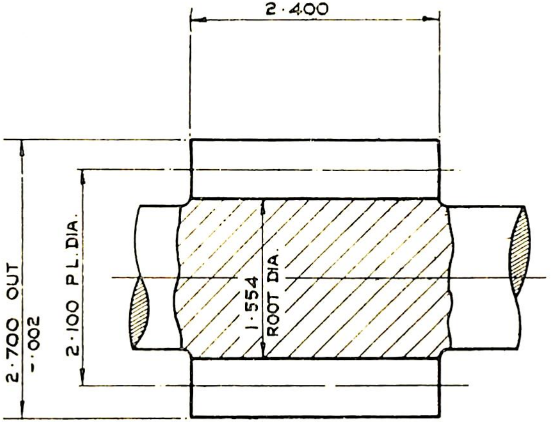
This diameter is generally so large that for all practical purposes it can be regarded as a straight line, but if however it is desired to calculate the corrections, they can be obtained in the same manner as described for spiral gears.

Example.

Given below are the calculations for a pair of module system gears 4/33 ratio to gear at 6·000 centres, the data obtained being neatly tabulated for both worm and wheel as shown in Figs. 6 and 7, the calculations for the worm being given first.

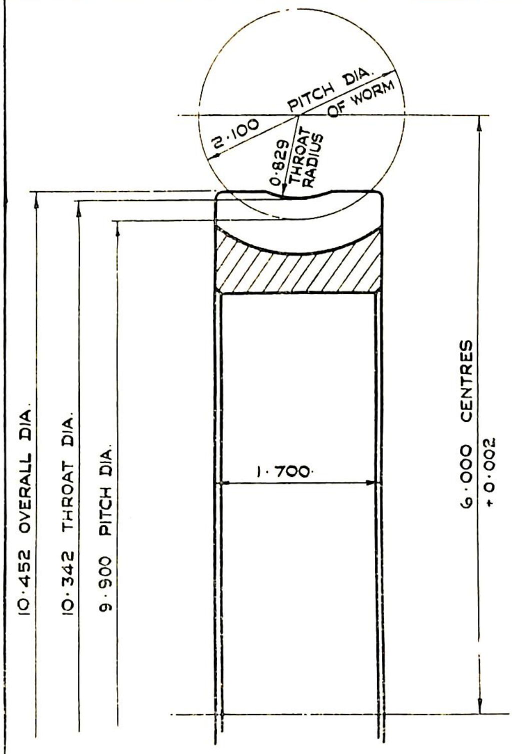
Then $M = \dfrac{2C}{T + q} = \dfrac{2 \times 6{\cdot}000}{33 + 7} = 0{\cdot}300$

Designation = 4/7/0·300R
$\tan \theta = t/q = 4/7 = 0{\cdot}57142 = 29° - 45'$
$a = M = 0{\cdot}300$
$b = M\,(2{\cdot}2 \cos \theta - 1) = 0{\cdot}300\,(2{\cdot}2 \times 0{\cdot}86820 - 1) = 0{\cdot}273$
$c = 0{\cdot}2\ M \cos \theta = 0{\cdot}2 \times 0{\cdot}300 \times 0{\cdot}86820 = 0{\cdot}052$
$d = q\,M = 7 \times 0{\cdot}300 = 2{\cdot}100$
$j = d + 2a = 2{\cdot}100 + 2 \times 0{\cdot}300 = 2{\cdot}700$
$i = d - 2b = 2{\cdot}100 - 2 \times 0{\cdot}273 = 1{\cdot}554$



CUTTING PARTICULARS	
TYPE OF GEAR	WORM
HAND	RIGHT
NUMBER OF THREADS	4
DIAMETER QUOTIENT	7
MODULE	0·300
AXIAL PITCH	0·9424
LEAD	3·7699
NORMAL PRESSURE ANGLE	21°-05′
HELIX OR LEAD ANGLE	29°-45′
BASE LEAD ANGLE	35°-54′
BASE DIAMETER	1·658
ADDENDUM	0·300
DEDENDUM	0·273
CHORDAL ADDENDUM	0·2865
CHORDAL THICKNESS	$0{\cdot}410^{-0{\cdot}008}_{-0{\cdot}012}$
BACKLASH	0·008/0·012
RUNNING CENTRES	6·000
CUTTER NUMBER	—
DESIGNATION	4/7/0·300
TO GEAR WITH WORMWHEEL DRG: Nº W 107	

FIG 6-WORKING DRAWING OF WORM.



CUTTING PARTICULARS	
TYPE OF GEAR	WORMWHEEL
HAND	RIGHT
NUMBER OF TEETH	31
ADDENDUM	0·221
DEDENDUM	0·352
RUNNING CENTRES	6·000
DESIGNATION	4/31/7/0·300
HOB NUMBER	—

TO GEAR WITH WORM :-
4 THREADS, RIGHT HAND,
0·300 ADDENDUM, 0·273
DEDENDUM, AT 6·000 CENTRES
DRG. Nº W.106.

FIG 7 WORKING DRAWING OF WORMWHEEL

At this stage the base diameter should be calculated on the basis of a normal pressure angle of 20°, and this diameter should not exceed $(i + 2c) = 1{\cdot}554 + 2 \times 0{\cdot}052 = 1{\cdot}658$

$$\tan \psi_t = \frac{\tan \psi_n}{\sin \theta} = \frac{\tan 20^\circ}{\sin 29^\circ{-}45'} = \frac{0{\cdot}36397}{0{\cdot}49622} = 0{\cdot}73348$$

$$= 36^\circ - 15'$$

$$d_o = d \cos \psi_t = 2{\cdot}100 \times 0{\cdot}80644$$

$$= 1{\cdot}693$$

In this case the base diameter d_o exceeds $(i + 2c)$ therefore d_o must be made equal to $(i + 2c)$.

$$\therefore \; d_o = 1{\cdot}658$$

$$\cos \psi_t = \frac{d_o}{d} = \frac{1{\cdot}658}{2{\cdot}100} = 0{\cdot}78952$$

$$= 37^\circ - 51'$$

$$\tan \psi_n = \tan \psi_t \sin \theta = 0{\cdot}77708 \times 0{\cdot}49622$$

$$= 0{\cdot}38560 \doteq 21^\circ{-}05'$$

As a check on the crest and root widths

$$\cos \psi_1 = \frac{d_o}{j} = \frac{1{\cdot}658}{2{\cdot}700} = 0{\cdot}61481$$

$$= 52^\circ - 03'$$

$\frac{1}{2}\,(\pi - 1{\cdot}25)$ M should be greater than $\dfrac{\text{L inv. } \psi_1}{2\,\pi}$

when $\text{L} = \text{M}\,\pi\,t$

inv. ψ_1 = involute function of angle ψ_1 from tables

$\frac{1}{2}\,(3{\cdot}1416 - 1{\cdot}25)\;0{\cdot}3$ should be greater than

$$\frac{0{\cdot}300 \times 3{\cdot}1416 \times 4 \times 0{\cdot}37380}{2 \times 3{\cdot}1416}$$

or $0{\cdot}28374$ should be greater than $0{\cdot}22428$ and in this case the sum of the crest and root widths will not be less than 1·25 M.

$$\cos \theta_o = \cos \theta \cos \psi_n = \cos 29^\circ{-}45' \cos 21^\circ{-}05'$$

$$= 0{\cdot}86820 \times 0{\cdot}93306$$

$$= 0{\cdot}81008$$

$$= 35^\circ - 54'$$

$$\text{F}_p = 8\,\text{M} = 8 \times 0{\cdot}300$$

$$= 2{\cdot}400$$

Vernier Settings of Worm.

$$d_m = i + \frac{(j - i)}{2} = 1{\cdot}554 + \frac{(2{\cdot}700 - 1{\cdot}554)}{2}$$

$$= 2{\cdot}127$$

$$\begin{aligned}
\tan\theta_m &= \frac{t\,M}{d_m} = \frac{4\times 0{\cdot}300}{2{\cdot}127} \\
&= 0{\cdot}56417 \\
&= 29^\circ - 26'
\end{aligned}$$

$$\begin{aligned}
t_c &= \frac{\pi M}{2}\cos\theta_m = \frac{3{\cdot}1416\times 0{\cdot}300}{2}\times 0{\cdot}87093 \\
&= 0{\cdot}410 \begin{array}{l} -\,0{\cdot}008 \\ -\,0{\cdot}012 \end{array}
\end{aligned}$$

$$\begin{aligned}
a_c &= \frac{j - d_m}{2} = \frac{2{\cdot}700 - 2{\cdot}127}{2} \\
&= 0{\cdot}2865
\end{aligned}$$

Wheel Calculations.

$$\begin{aligned}
A &= M\,(2\cos\theta - 1) = 0{\cdot}300\,(2\times 0{\cdot}86820 - 1) \\
&= 0{\cdot}221 \\
\text{or } A &= b - c = 0{\cdot}273 - 0{\cdot}052 \\
&= 0{\cdot}221 \\
B &= M\,(0{\cdot}2\cos\theta + 1) = 0{\cdot}300\,(0{\cdot}2\times 0{\cdot}86820 + 1) \\
&= 0{\cdot}352 \\
\text{or } B &= a + 0{\cdot}052 = 0{\cdot}352 \\
D &= 2C - q\,M = 2\times 6{\cdot}000 - 7\times 0{\cdot}300 \\
&= 12{\cdot}00 - 2{\cdot}100 \\
&= 9{\cdot}900
\end{aligned}$$

$$\begin{aligned}
J &= D + 2A = 9{\cdot}900 + 2\times 0{\cdot}221 \\
&= 10{\cdot}342 \\
I &= D - 2B = 9{\cdot}900 - 2\times 0{\cdot}342 \\
&= 9{\cdot}196 \\
O &= D + 2{\cdot}5A = 9{\cdot}900 + 2{\cdot}5\times 0{\cdot}221 \\
&= 10{\cdot}452 \\
F_w &= 2M\sqrt{q+1} = 2\times 0{\cdot}300\sqrt{7+1} \\
&= 0{\cdot}600\times 2{\cdot}828 \\
&= 1{\cdot}6968 \text{ make } 1{\cdot}700 \\
G &= C - J/2 = 6{\cdot}000 - 5{\cdot}171 \\
&= 0{\cdot}829
\end{aligned}$$

CHAPTER II.

WORM GEAR MANUFACTURE.

Worm gear manufacture should be considered in two parts :—

(1) the worm.
(2) the wormwheel,

there being a distinct difference between the two, as far as the material used, and the method of manufacture is concerned. Many combinations of materials may have been used in the past, but where continuous power transmission at relatively high speeds is required, the choice of materials can be narrowed down to hardened steel for the worm, and phosphor-bronze for the wheel. In first considering the production of the worm threads, however, three methods are used for cutting, in addition to grinding and polishing after hardening. The most common method is to use a disc type milling cutter (usually cam relieved) one thread being milled at a time.

In addition, worms can be hobbed, in which case the worm is treated as a spiral gear, or alternatively the threads can be cut with a pinion type cutter, similar to the cutters used for spur gears produced by the gear shaper process. Pinion type cutters should only be used, however, for worms of low lead angles and when the thread form is straight sided on the axial section, *e.g.*, for the ordinary type worms of low lead angle or for acme screw threads.

When hobs are used for cutting worms the hob is made the same normal pitch as the worm to be cut, and the same basic normal pressure angle ; the hob then generates the correct tooth shape on the axial, and transverse planes as specified on the drawing.

This method is only recommended for worms having very large helix angles, and which are in fact equivalent to spiral gears of special tooth proportions. For general worm production, however, it will be found that the milling method fulfils all requirements, and for this operation worm milling machines are available. The worm blank is held between centres or in a collect chuck, and roller steady, and is caused to advance spirally past a rotating cutter which is carried in an adjustable cutter head, the head being adjusted to an angle equal to the helix angle of the worm being cut.

During this movement a thread space is milled, the feed being tripped on completion, after which the blank is returned to the starting point, indexed according to the number of threads to be cut, and the cycle repeated.

The cutter form must correspond to the worm tooth space on the normal section, and for module system worms this form is obtained by generating a lead blank in the worm milling machine.

The lead blank conforms in every way to the size of the cutter desired and is mounted on the cutter spindle of the worm milling machine, the cutter head being set to the lead angle of the worm. In place of the worm, an arbor is used on which is secured a cutter blade, and when the arbor is traversed in a spiral of the desired lead it sweeps out a surface identical with that of the corresponding worm.

The lead blank is then rotated at the cutting speed, and the cutting edge on the arbor traversed slowly past it with the result that the correct cutter profile on the lead blank is obtained. Gauges are then made from the shape produced for use when manufacturing the actual high speed steel cutter used to cut the worm threads.

Another method is to first produce a master worm on the centre lathe or worm thread grinding machine from which the cutter gauges are made on a plane normal to the lead angle of the worm.

When chasing the worm threads in the centre lathe the angle of the tool used is made equal to the base lead angle, while the top cutting edge of the tool is disposed above the axis of the worm by an amount equal to half the base circle radius.

The normal chordal thickness of the cutter is made less than the finished normal chordal space width of the worm, if it is desired to grind the worm thread profiles, this allowance varying with the pitch of the worm, but 0·006 to 0·010 on each flank will be found satisfactory.

For worms having large leads it is customary to mill the worm threads at a slightly shorter lead than that given on the finished drawing to allow for distortion in hardening, the amount by which the lead is shortened being 0·001 per inch of lead.

After carburising and hardening, the outside diameter of the worm is ground finished drawing size, to a tolerance of at least 0·003, and the worm threads ground on a worm thread grinding machine.

This machine is very similar to the worm milling machine, a grinding wheel being used in place of the worm milling cutter, and an additional swivel movement provided for the wheel head.

The flat side of the wheel is used for module system worms, the wheel head swivelled round to the base lead angle, and the wheel spindle set to the helix angle of the worm.

The flat side of the wheel is set in such a position that the flank of the worm thread is ground straight sided on the base cylinder of the worm, but a slight adjustment to the base angle setting is sometimes necessary to obtain the correct thread form.

Change gears are provided to obtain the desired lead, and number of threads, on both milling and grinding machines, the formula for the change gear ratio being usually indicated on the change gear covers, although full instructions are given in the instruction book supplied with each machine.

Wormwheels.

Wormwheel teeth are cut on a specially designed hobbing machine using a full hob for large production or a fly hob for a small lot production.

In both cases the hob tooth form is a facsimile of the worm tooth, the hob tooth being made a longer addendum to cut the clearance and a shorter dedendum if it is desired to cut the throat or gorge radius in the wormwheel blank when using a full hob.

Two cutting methods can be used, namely :—

(*a*) Radial feed.
(*b*) Axial feed,

the radial feed method as shown in Fig. 8 should only be used for wormwheels up to 6 to 8 degrees lead angle, or when the gear tooth form is not of extreme importance. In either case the hob should be set at the finishing operation, to the same centre distance as the worm and wheel when assembled, providing that the hob used is the same pitch circle diameter as the worm. When a new hob is used, however, this is made about 0·030 greater than the

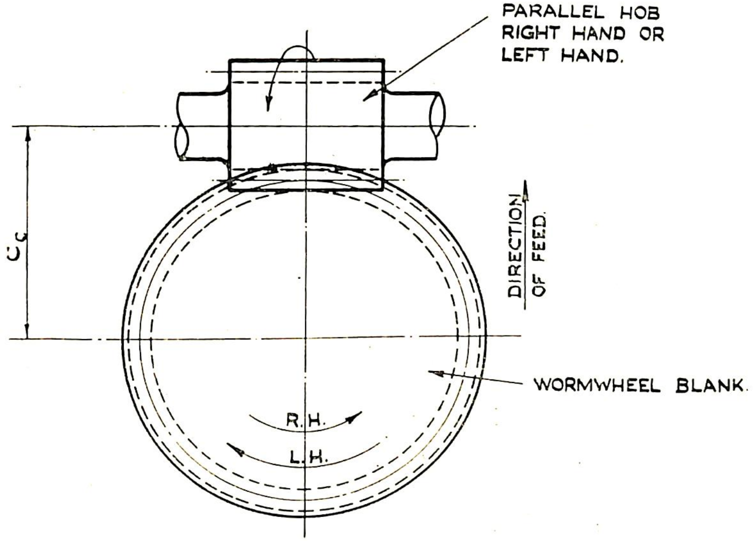


FIG. 8. RADIAL FEED METHOD OF CUTTING WORMWHEEL BLANK

worm, in this case the hob should be measured, and half the increase over nominal being added to the running centres to obtain the cutting centres.

On some machines this increase in hob diameter is compensated for by a slight adjustment to the helix angle, therefore it is advisable to check the outside diameter and re-calculate the lead angle before using, or each time the hob has been re-sharpened, the calculations being made as follows :—

let O_H = outside diameter of hob
D_H = pitch diameter of hob
D = pitch diameter of worm
C = running centres of worm and wheel
C_C = cutting centres
B = dedendum of wheel and hob addendum
θ_H = lead angle of hob
L = lead of worm

then after measuring hob to obtain value O_H we have

$$D_H = O_H - 2B$$

and

$$\tan \theta_H = \frac{L}{\pi D_H}$$

then

$$C_C = C + \frac{D_H - D}{2}$$

A more accurate wormwheel tooth form can be obtained by using the axial or tangential feed method, as there is then no possible chance of cutting away part of the gear tooth flanks which are required on the finished tooth form.

The hob is set at the correct cutting centres, and offset to the centre of the machine table as shown in Fig. 9, and fed past the blank in an axial direction until the hob teeth clear the wheel tooth flanks.

This method can be used for wormwheels of any lead angle, and is especially suitable when using fly hobs.

Fly hobs differ from full hobs in that all but one or a few teeth are mounted as separate cutters in a suitable arbor.

Fly hobs used for finishing need have only one cutter, but for a normal wormwheel meshing with a multiple thread worm it will be necessary to index the hob at the end of each traverse.

The axial feed method has a great advantage over the radial feed method, in so far that after each revolution of the tool a different set of hob teeth is being used to cut the tooth flanks of the wheel.

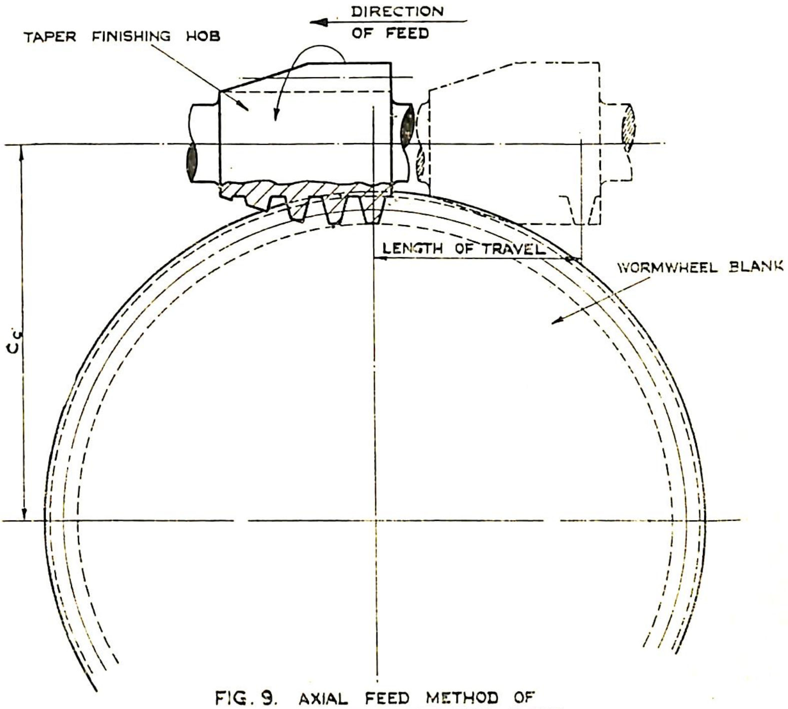


FIG. 9. AXIAL FEED METHOD OF CUTTING WORMWHEEL BLANK.

This means that the number of tooth positions which are used to make up the tooth profile of the wormwheel, depend only upon the rate of feed, and not upon the number of flutes or starts in the hob.

Worm wheel blanks should be accurately turned, and located for cutting in the same bore or registers used for mounting in the gear casings in which they are to be used.

Care should also be taken to cut the wormwheel teeth in correct relation to the end locating face, and in some cases it may be necessary to machine the blanks to closer limits than specified on the finished drawing, for this purpose.

In other words the hob axis should be set when cutting, to the same position in relation to the end locating face as the worm axis when assembled in the casing, and when a machine is being used without any vertical adjustment on the hob spindle housing the fixture used for cutting has to be made a definite height from the base to the locating face.

To overcome this difficulty for small lot production an adjustable fixture is sometimes used, but it is found in practice that these

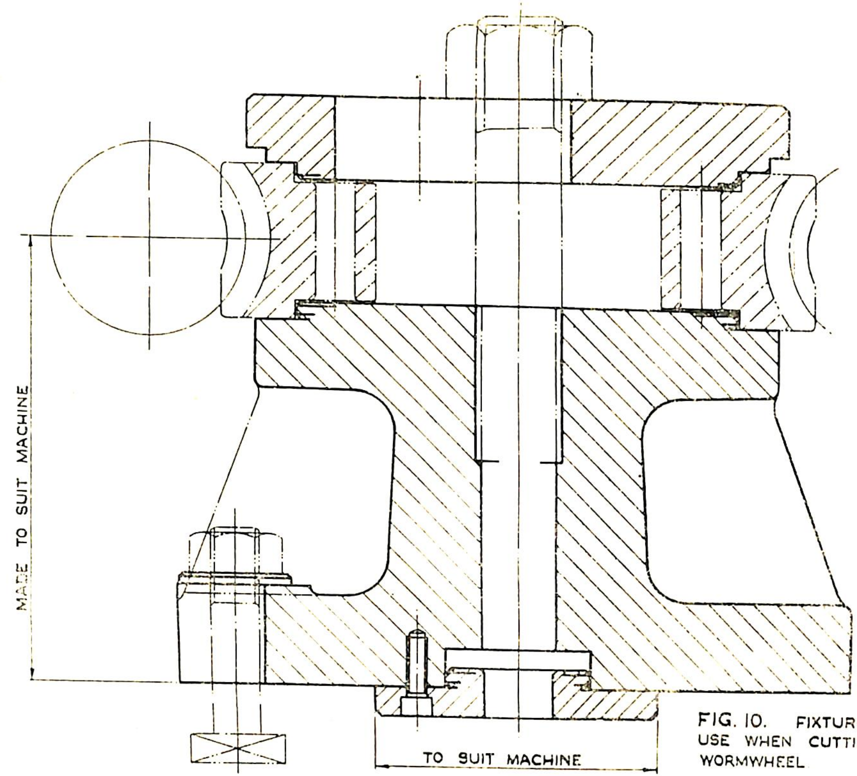


FIG. 10. FIXTURE FOR USE WHEN CUTTING WORMWHEEL

are not quite as rigid as solid fixtures, and it is more economical in the long run to use solid fixtures generally as shown in Fig. 10.

In any case it is advisable to make a capacity layout chart for each machine installed for the purpose of designing suitable fixtures without having to refer, or measure, the machine each time a fixture is required.

Should it be necessary to cut a wormwheel which is required to gear with a worm in which the number of teeth in the wheel and the number of threads in the worm will factorize, it is customary to use a hob having one or more threads, than the worm.

For example, taking 12 x 36 ratio we should make the hob equivalent to 13 threads and as this increases the pitch diameter of the hob we have

$$D_H = \frac{D \times t_H}{t}$$

$$L_H = p_a \times t_H$$

when D_H = pitch diameter of hob having t_H threads
D = pitch diameter of worm having t threads
t_H = number of threads in hob
t = number of threads in worm
L_H = lead of hob
p_a = axial pitch of worm

It will be found that the helix angle θ of the hob is the same as the worm, but as the hob is greater in diameter than the worm, the bedding will favour the centre of the wormwheel face in the initial stages.

It is preferable, however, when designing worm gears to have no common factors between the number of teeth in the wheel and the number of threads in the worm.

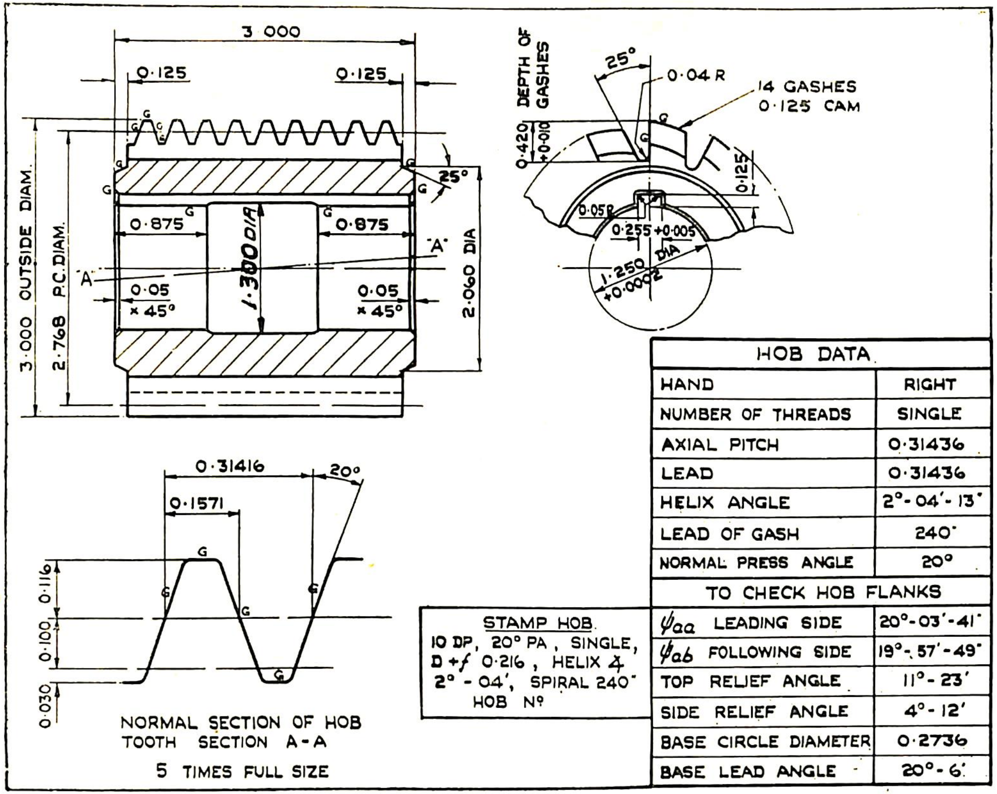
CHAPTER III.

HOB DESIGN FOR SPUR, SPIRAL AND HELICAL GEARS.

Before proceeding with the design of any hob, it is essential that full particulars are available of the machine to be used, and of the teeth to be cut.

The machine details can be obtained from the machine layout drawings previously mentioned or the specification supplied with the machine, and from this layout the following information is required :—

(*a*) maximum hob diameter
(*b*) diameter and keyway sizes of arbor used.
(*c*) hob spindle sizes, if a shank hob is being used.



HOB DATA	
HAND	RIGHT
NUMBER OF THREADS	SINGLE
AXIAL PITCH	0·31436
LEAD	0·31436
HELIX ANGLE	2°-04′-13″
LEAD OF GASH	240″
NORMAL PRESS ANGLE	20°
TO CHECK HOB FLANKS	
ψ_{aa} LEADING SIDE	20°-03′-41″
ψ_{ab} FOLLOWING SIDE	19°-57′-49″
TOP RELIEF ANGLE	11°-23′
SIDE RELIEF ANGLE	4°-12′
BASE CIRCLE DIAMETER	0·2736
BASE LEAD ANGLE	20°-6′

STAMP HOB.
10 DP, 20° PA, SINGLE,
D + f 0·216, HELIX ∠
2° - 04′, SPIRAL 240″
HOB Nº

FIG. 11. WORKING DRAWING OF SPUR GEAR HOB

With regard to the tooth particulars, these may be taken from the table shown on the working drawing or if the gear to be cut has teeth of standard proportions the particulars can easily be calculated or taken from tables.

It is preferable, however, to use the working drawing in every case, and thus enable the designer to check any clearances should it be necessary, especially in the case of cluster spur gears, or gears having shoulders, etc.

The length of the hob is not important providing it is not too short, a good rule being to fix a minimum of two pitches and a maximum of eight pitches, the cutting portion together with the length of the collars at each end, fixing the overall length of the hob.

A typical working drawing for a spur gear hob is shown in Fig. 11, and it will be noted that the cutting portion is made parallel along the whole length.

The same hob can be used for cutting spiral or helical gears providing that the normal pitch and normal pressure angle of the hob is the same as the normal pitch and normal pressure angle of the gear.

For large production of spiral or helical gears, however, it is advisable to introduce a slight taper on the leading end of the hob, and as the hob should also be made the same hand as the gear, two hobs are necessary for left and right hand gears.

The leading ends for right and left hand hobs are opposite in relation to the direction of cut generally as shown in Figs. 12 and 13, Fig, 12 showing a right hand hob for right hand gears, and Fig. 13 a left hand hob for left hand gears.

The taper at the leading end is usually made about 15° to 20° the depth of the taper being approximately equal to the cutting depth of the hob tooth.

The reason for the taper is that a stronger tooth is presented to the work at the commencement of the cut, thus preventing the hob teeth from being broken.

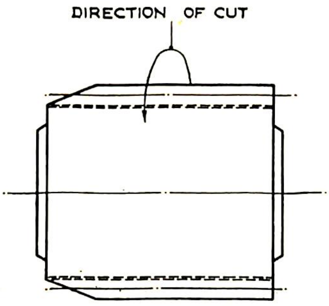


FIG. 12. RIGHT HAND HOB FOR RIGHT HAND GEARS.

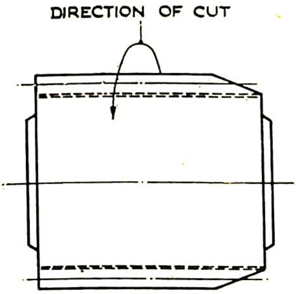


FIG. 13. LEFT HAND HOB FOR LEFT HAND GEARS.

Finishing Hobs.

Finishing hobs should be made as large in diameter as the machine and component will allow, and they should also be made single thread.

Using a single thread hob, one ring of teeth will completely generate one gear tooth, and as the hob is provided with gashes each gash is presented in a different position relative to the gear tooth, thus completely generating the involute tooth form.

To obtain a clear picture of the hob and its function, it should be imagined that the hob is a worm in mesh with a spur gear, and that the hob is swung over, normal to the teeth of the spur gear.

That being the case the hob is made the same normal pitch as the spur gear, and as the normal pitch of the gear is equivalent to the circular or transverse pitch the normal pitch of the hob should be identical.

The same rule applies to the pressure angle of the hob, namely, the normal pressure angle of the hob should equal the normal or transverse pressure angle of the gear to be cut.

For spiral or helical gears the normal pressure angles of both hob and gear should be identical, although in some instances both for spur and spiral gears the hob can be made a different normal pitch to the gear.

In any event the relative values of the hob and gear should satisfy the following equation :—

$$p_n \cos \psi_n = p_{nc} \cos \psi_{nc}$$

when p_n = normal or transverse pitch of gear
ψ_n = normal or transverse pressure angle of gear
p_{nc} = normal pitch of hob
ψ_{nc} = normal pressure angle of hob or cutter

and it should be specially noted that this formula can be applied to any involute generating cutter.

In manufacturing the hob, however, the axial pitch p_{ac} is used, or in the case of multiple thread hobs, the lead, and this value depends upon the helix angle of the hob threads.

It is therefore essential to layout the end view of the proposed hob as shown in Fig. 14 before any calculations affecting the helix angle and axial pitch, are made.

Assuming that the outside diameter and size of bore have been fixed, the full depth of the hob tooth is set off from the outside diameter and a trial gash set out in the manner shown.

It is advisable to have as many gashes as possible without making the hob teeth too weak, and after deciding upon the number of gashes, the cam used for relieving the hob teeth can be obtained by setting out a small triangle as shown in figure.

The cam used should be such that the angle or top rake is about 10° and if the distance G is obtained from the end view layout of the hob, the nearest cam available can easily be determined.

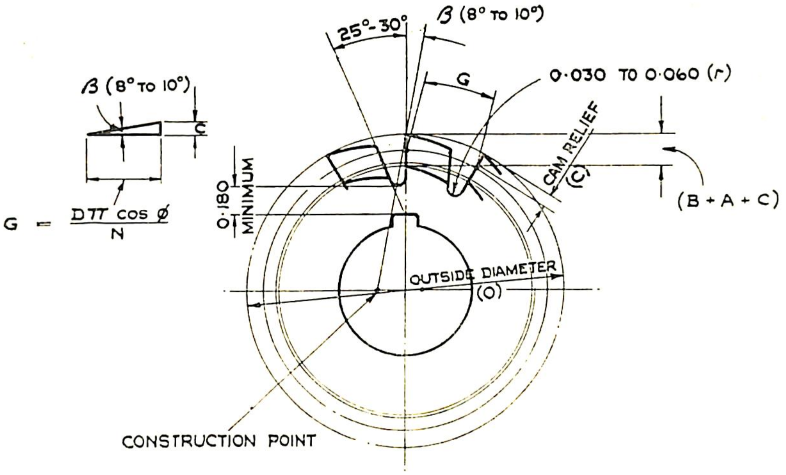


FIG. 14 LAYOUT OF GASHES FOR SPUR GEAR HOB

After fixing the number of gashes and the cam relief, the distance from the root of the gash to the keyway (if a bored hob is being used) should not be less than 0·180 depending of course on the size of the hob. The value given would be suitable for a hob 10 DP pitch, and if it is not possible to obtain this minimum, then a solid type shank hob will be necessary.

The depth of gash should also be specified and this can be obtained from the layout or calculated.

If calculated, the gash depth will be equal to

$$B + A + c + C + r$$

when B = hob addendum or gear dedendum
A = hob dedendum or gear addendum
c = hob cutting clearance
C = cam relief
r = radius of gashing cutter

the cutting depth of the hob being equal to $(B + A)$.

Tooth Form.

Theoretically the normal section of the hob tooth should be slightly convex, the exact shape being obtained from the involute on the end of transverse plane of the hob, and checked by the same method as used for a module system involute worm as described in Chapter I.

The involute on the transverse plane is not apparent however, due to the small helix angles employed for spur gear hobs, and is further complicated by the presence of the gashes, and cam relief.

As a compromise, and also to assist manufacture, the hob tooth on the normal plane Section A.A., Fig. 11, is made with straight sides, and providing that the hob is made as large as possible in diameter, and single thread, thereby keeping the helix angle as low as possible, the slight error in hob tooth shape is of no account.

To obtain the necessary particulars for manufacturing the hob we proceed as follows :—

Given O
Then $D = O - 2B$
when O = outside diameter of hob
D = pitch diameter of hob
B = addendum of hob or dedendum of gear

and $\sin \phi = \dfrac{p_n t}{D \pi}$

when ϕ = helix angle of hob
p_n = normal pitch of hob in inches
t = number of threads

Note.—Given the pitch in terms of diametral pitch for a spur gear, or the normal diametral pitch for a spiral or helical gear,

$$p_n = \frac{\pi}{DP} \text{ inches}$$

from which

$$p_a = \frac{p_n}{\cos \phi}$$

when p_a = axial pitch of hob
and $L = p_a t$
when L = lead of hob

The gashes are produced at right angles to the hob thread in the majority of cases and it is therefore necessary to specify the lead of gash, or spiral.

This lead is used on the hob sharpening machine each time the hob is sharpened and the value should therefore be etched on the hob. For hobs under one degree helix angle it would be in order to make the hob with straight gashes, *i.e.*, parallel to the hob axis.

Let S = lead of gashes or spiral
then $S = D \pi \cot \phi$
and $\theta = 90^\circ - \phi$
when θ = gash angle

The amount of top relief can be calculated when the size of cam to be used is known and from the top relief the side relief can be obtained and although this value is not used for manufacture it is advisable to know how much side relief is present, should the hob show any signs of rubbing on the flanks.

Then $\tan \beta = \dfrac{C\,N}{D\,\pi \cos \phi}$

when β = top relief angle
C = cam relief in inches
N = number of gashes
D = pitch diameter of hob
ϕ = helix angle of hob

and $\tan \alpha = \tan \psi_n \tan \beta$

when α = side relief angle
ψ_n = normal pressure angle
β = top relief angle

To check the hob tooth flanks on the assumption that they have straight sides on the normal plane, *i.e.*, for finishing hobs having small helix angles, it is necessary to calculate the axial pressure angle ψ_a

$$\text{when } \tan \psi_a = \frac{\tan \psi_n}{\cos \phi}$$

This angle is correct when no relief has been machined on the hob teeth, and due to this relief the axial pressure angle is slightly different on each side of the hob tooth, being greater on the leading side and less on the following side by equal amounts.

Thus :

$$\tan \psi_{aa} = \frac{\sec \phi}{\cot \psi_n - \tan \beta \tan \phi}$$

$$\text{and } \tan \psi_{ab} = \frac{\sec \phi}{\cot \psi_n + \tan \beta \tan \phi}$$

when ψ_{aa} = axial pressure angle leading side
ψ_{ab} = axial pressure angle following side
ϕ = helix angle of hob
ψ_n = normal pressure angle
β = top relief angle

To measure the hob tooth flanks in the same manner as used for an involute worm it is necessary to calculate the base diameter and base lead angle of the hob and referring to Fig. 15.

$\cos \phi_b = \cos \psi_n \cos \phi$

when ϕ_b = base lead angle or the helix angle on the base diameter of the hob
ψ_n = normal pressure angle
ϕ = helix angle of hob

from which the base diameter of the hob is equal to

$$D_o = \frac{L}{\tan \phi_b\, \pi}$$

when D_o = base diameter of hob
L = lead of hob
ϕ_b = base lead angle

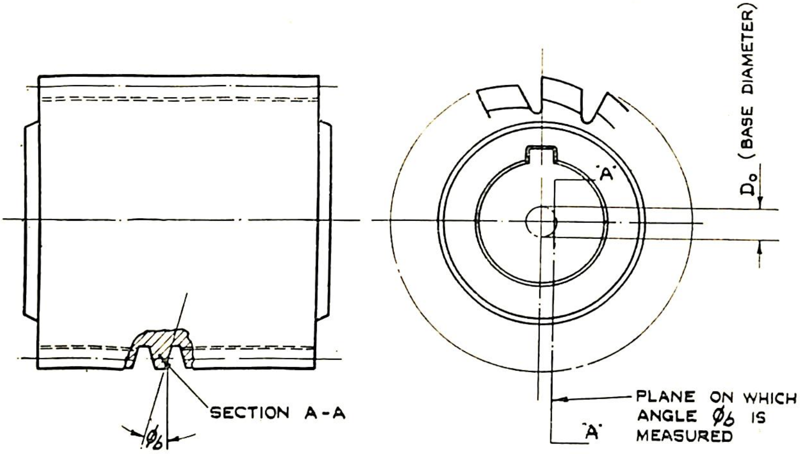


FIG. 15. MEASURING HOB TOOTH FLANK USING BASE DIAMETER AND BASE LEAD ANGLE.

Tip Relief.

On finishing hobs it is sometimes found necessary to design the hob tooth flanks to remove the tips of the gear tooth to be cut.

This is termed tip relief, and it is usually expressed in terms of the distance down the flank (*a*) and the amount (*b*) to be removed from the extreme tip of the gear tooth as shown in Fig. 16.

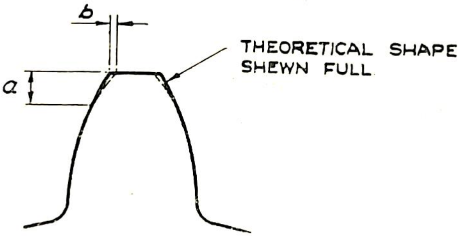


FIG. 16. TIP RELIEF ON GEAR TOOTH

Tip relief should, however, be avoided wherever possible as it is very expensive, due to the fact that different hobs are required for gears having different numbers of teeth, even when the gears are the same pitch and pressure angle.

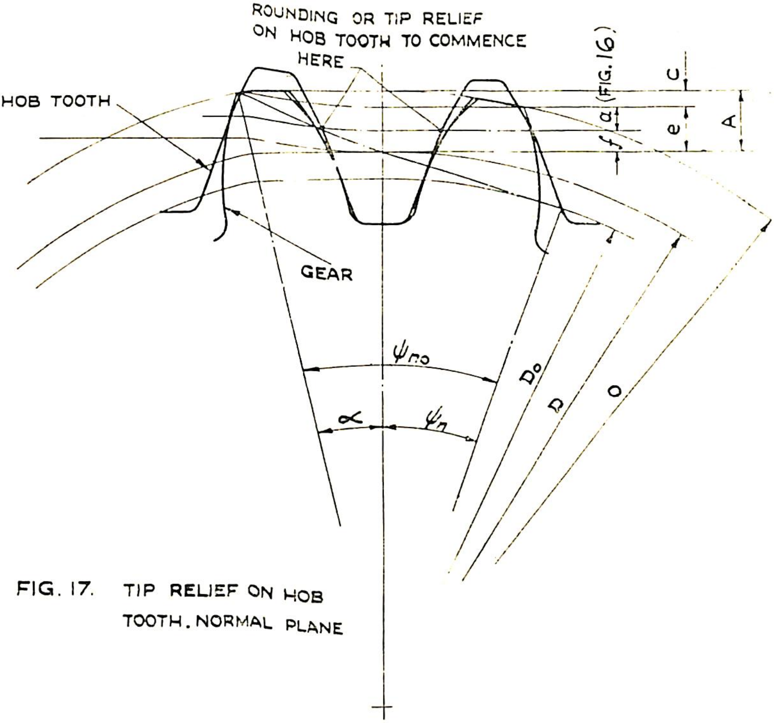


FIG. 17. TIP RELIEF ON HOB TOOTH. NORMAL PLANE

To provide definite tip relief on the hob, the hob tooth is thickened up in the roots and referring to Fig. 17

ψ_n = normal pressure angle of gear to be cut
ψ_{no} = pressure angle of gear at the outside diameter O.
D = pitch circle diameter of gear
O = outside diameter of gear
A = addendum of gear
f = distance from hob pitch line to commencement of tip relief on hob tooth

$$\cos \psi_{no} = \frac{D \cos \psi_n}{O}$$

$$\alpha = \psi_{no} - \psi_n$$

distance c = $O/2\,(1 - \cos \alpha)$
distance e = $A - c$
then distance $f = e - a$ (referring to Fig. 16)

The above method can be used for rack generating cutters for spur gears in addition to hobs, and can also be used for spiral and helical gears substituting D by the virtual pitch circle diameter D_v.

Example.

Based on the foregoing, calculations have been made for a spur gear hob suitable for cutting a gear 10 DP pitch 20° pressure angle.

From the layout of the machine selected we can use a hob 3·000 outside diameter, 3·000 long with 1·250 bore.

$$\begin{aligned}
\text{Then}\quad O &= 3{\cdot}000\\
p_n &= 10\ \text{DP}\ (0{\cdot}31416)\\
\psi_n &= 20^\circ\\
\text{and}\quad D &= 3{\cdot}000 - 2 \times 0{\cdot}116\\
&= 2{\cdot}768
\end{aligned}$$

$$\begin{aligned}
\sin\quad \phi &= \frac{0{\cdot}31416 \times 1}{2{\cdot}768 \times \pi}\\
&= 2^\circ - 04' - 13''
\end{aligned}$$

$$\begin{aligned}
p_a &= \frac{0{\cdot}31416}{\cos 2^\circ - 04' - 13''}\\
&= 0{\cdot}31436\\
L &= 0{\cdot}31436 \times 1\\
&= 0{\cdot}31436\\
S &= 2{\cdot}768 \times 3{\cdot}1416 \times \cot 2^\circ - 04' - 13''\\
&= 240{\cdot}98\\
&\quad \text{say 240 inches}\\
\theta &= 90^\circ - 2^\circ - 04' - 13''\\
&= 87^\circ - 55' - 47''
\end{aligned}$$

$$\begin{aligned}
\tan\quad \beta &= \frac{0{\cdot}125 \times 14}{2{\cdot}768 \times 3{\cdot}1416 \times \cos 2^\circ - 04' - 13''}\\
&= 0{\cdot}20138\\
&= 11^\circ - 23'\\
\tan\quad \propto &= \tan 20^\circ \times \tan 11^\circ - 23'\\
&= 0{\cdot}36397 \times 0{\cdot}20138\\
&= 0{\cdot}07320\\
&= 4^\circ - 12'
\end{aligned}$$

$$\begin{aligned}
\tan\quad \psi_a &= \frac{\tan 20^\circ}{\cos 2^\circ - 04' - 13''}\\
&= 20^\circ = 00' - 41''
\end{aligned}$$

$$\begin{aligned}
\tan\quad \psi_{aa} &= \frac{\sec 2^\circ - 04' - 13''}{\cot 20^\circ - \tan 11^\circ\ 23' \times \tan 2^\circ - 04' - 13''}\\
&= \frac{1{\cdot}0007}{2{\cdot}7475 - 0{\cdot}00727}\\
&= 0{\cdot}36519\\
&= 20^\circ - 03' - 41''
\end{aligned}$$

$$\tan \psi_{ab} = \frac{1{\cdot}0007}{2{\cdot}7475 + 0{\cdot}00727}$$

$$= 0{\cdot}36325$$

$$= 19^\circ - 57' - 49''$$

$$\cos \phi_b = \cos 20^\circ \times \cos 2^\circ - 04' - 13''$$

$$= 0{\cdot}93907$$

$$= 20^\circ - 6' - 0$$

$$D_o = \frac{0{\cdot}31436}{\tan 20^\circ - 6' \times 3{\cdot}1416}$$

$$= 0{\cdot}2736$$

Roughing Hobs.

Roughing hobs should only be used for medium and coarse pitch gears, as in the case of fine pitch gears it is possible to complete the gear cutting operation in one cut using the finishing hob.

Roughing hobs are usually made with multiple threads to speed up production, a triple thread hob being used for medium pitches and a double thread hob for course pitches.

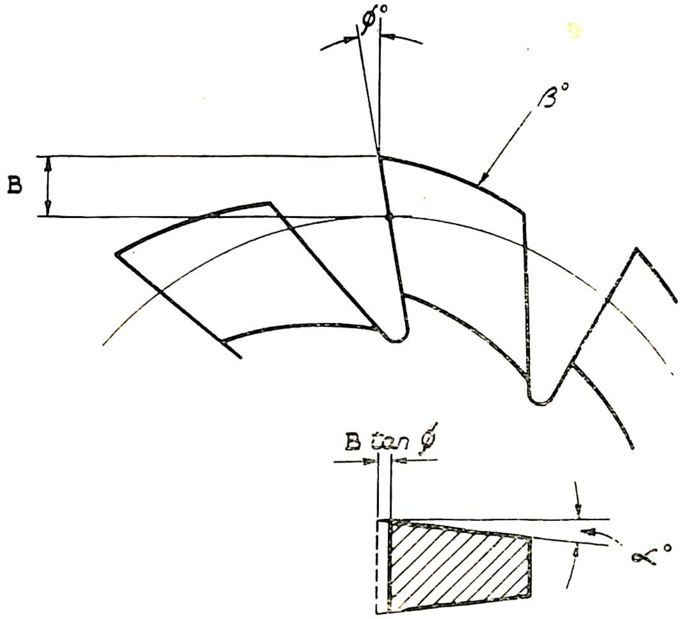


FIG. 18. SHEWING POSITIVE FRONT RAKE ON ROUGHING HOB.

These hobs do not generate a very accurate tooth profile especially if the tooth flanks on the normal plane are made with straight sides, but as an allowance has to be made for finishing, the slight error produced is thus eliminated.

The amount left on the circular tooth thickness of the gear varies of course with the pitch, but an allowance of 0·030 for 8 DP up to 0·060 for 2 DP gears will be found satisfactory.

The main difference between a roughing hob and a finishing hob apart from the points mentioned is that a positive front rake or relief of 5° to 10° is introduced, generally as shown in Fig. 18.

The angle of rake incorporated in the design is important, and the value in degrees should be engraved on the hob to ensure that it is sharpened correctly.

This is due to the fact that by introducing a positive front rake it necessitates a slight increase to the normal pressure angle of the hob ψ_n, and the increased value must be used for all subsequent calculations as used for finishing hobs.

The first step therefore after fixing the amount of front rake is to calculate the new value for the normal pressure angle of the hob, then

Let ψ_n = normal pressure angle of gear
ψ_{nh} = normal pressure angle of roughing hob
β = top relief angle of hob
ϕ = front relief angle of hob

$$\text{then } \tan \psi_{nh} = \tan \psi_n \left(\frac{\cos \beta \cos \phi}{\cos (\beta + \phi)} \right)$$

The introduction of front rake also complicates the measurement of the hob tooth with a pair of gear tooth verniers, and by referring again to Fig. 18

B = addendum of hob

then height setting of vernier measured on cutting edge will equal B sec ϕ, and the thickness setting will equal

nominal hob tooth thickness $- 2 (B \tan \phi \tan \alpha)$
when $\tan \alpha = \tan \psi_{nh} \tan \beta$
,, ,, α = side relief angle of hob

CHAPTER IV.

GEAR SHAPER CUTTER DESIGN FOR WORMS AND SCREWS.

Gear shaper cutters are used for cutting worms and acme threads of low lead angles on machines specially designed for the purpose.

Worms for automobile steering gearbox applications are specially suitable for this method of production, due to the fact that the worm threads are usually made straight sided on the axial plane.

To generate the shape, which is in fact the same shape as an acme thread, an involute form cutter is required, the only difference between this and a gear shaper cutter for spur gears being found in the thread proportions and relief angles of the cutter.

In Fig. 19 is shown a diagrammatic sketch of the type of machine used, the most important factor governing the design of the cutter being the wormwheel used in the drive to the cutter spindle. The cutter pitch circle diameter must in every case be the same as the pitch circle diameter of the wormwheel fitted to the machine, the standard wormwheel having 20 teeth of 0·500 circular pitch and a pitch circle diameter of 3·183 inches. With this wormwheel it is possible to cut the following axial pitches, and in the table is also specified the required number of teeth in the cutter.

Pitch of Cutter	0·100	0·125	0·200	0·250	0·3125	0·400	0·500
No. of Teeth in Cutter	100	80	50	40	32	25	20

Other axial pitches may be cut on these machines however, providing that a wormwheel of the same pitch circle diameter as the cutter is supplied, together with a suitable worm.

The top and front relief angles of cutters used for generating worm threads are 10° and 15° respectively, and to generate the correct axial pressure angle, the cutter pressure angle is modified, due to the above mentioned relief angles,

when ψ_a = axial pressure angle of work
ψ_{tc} = transverse pressure angle of cutter

$$\text{then } \tan \psi_{tc} = \tan \psi_a \left(\frac{\cos 10^\circ \cos 15^\circ}{\cos (10^\circ + 15^\circ)} \right)$$

Note.—when $\psi_a = 14^\circ - 30'$ $\psi_{tc} = 15^\circ - 11'$

A nominal amount of 0·050 is added to the front face of the cutter as a sharpening allowance from the true section of the cutter

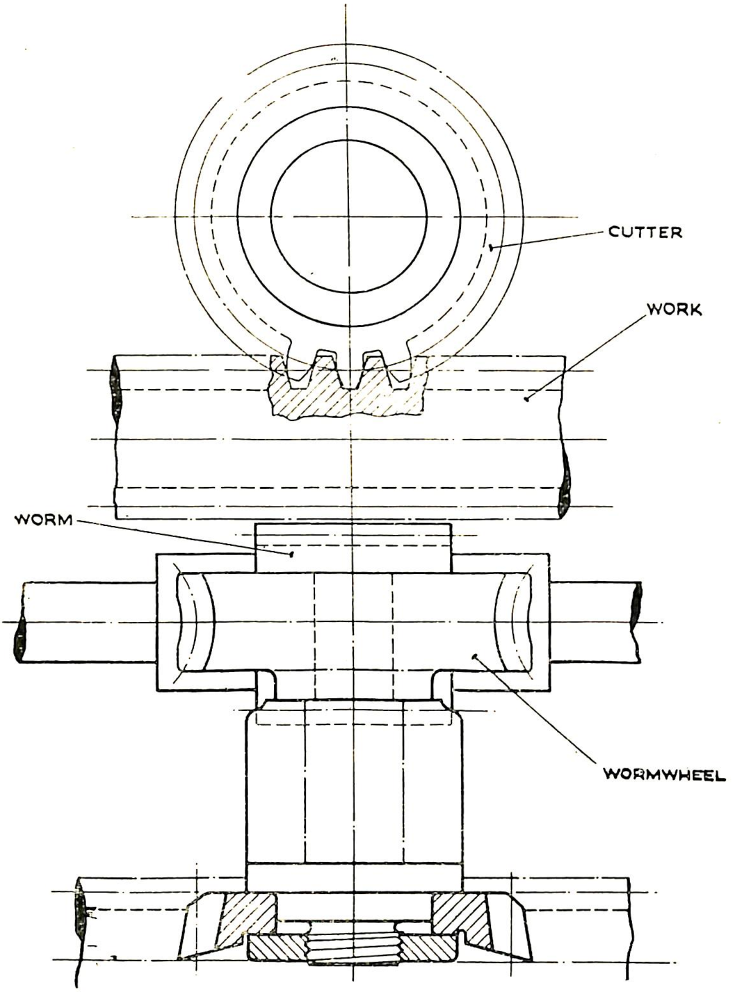


FIG. 19. DIAGRAMMATIC SKETCH OF THREAD GENERATING MACHINE CUTTER SPINDLE DRIVE

(this amount being included in the calculations), and when sharpening the cutter the front angle face of 15° is only ground, in the same manner as a spur gear shaper cutter.

Before designing a cutter of this type it is essential to have full particulars of the worm thread proportions on the axial plane as shown in Fig. 20, in addition to the usual specification layout of the machine to be used.

The cutter should be made the same hand as the threads to be cut, the cutter particulars being calculated as follows :—

when Z_c = number of teeth in cutter
D_c = pitch circle diameter of cutter
D_{oc} = base circle diameter of cutter
ψ_{tc} = pressure angle of cutter
ψ_a = axial pressure angle of worm threads
O_c = outside diameter of cutter
B_o = addendum of cutter on O_c
A = dedendum of cutter
a = addendum of worm threads
b = dedendum of worm threads
t_a = axial thickness of worm thread on d
d = pitch circle diameter of worm.
p_a = axial pitch of worm
y = sharpening allowance
t_c = circular thickness of cutter tooth
t_{cc} = normal chordal thickness of cutter tooth
B_{oc} = normal chordal addendum of cutter tooth
D_{cv} = virtual pitch circle diameter of cutter
t_{cn} = normal circular thickness of cutter tooth
θ_n = half angle subtended by t_{cn} on diameter D_{cv}
f = correction factor
L_c = lead of cutter
ϕ_c = spiral angle of cutter
t = number of threads in worm

then $O_c = D_c + 2\,(b + y \tan 10^\circ)$ when $y = 0{\cdot}050$

$t_c = (p_a - t_a) + 2\,(y \tan 10^\circ \tan \psi_{tc})$

$t_{cn} = t_c \cos \phi_c$

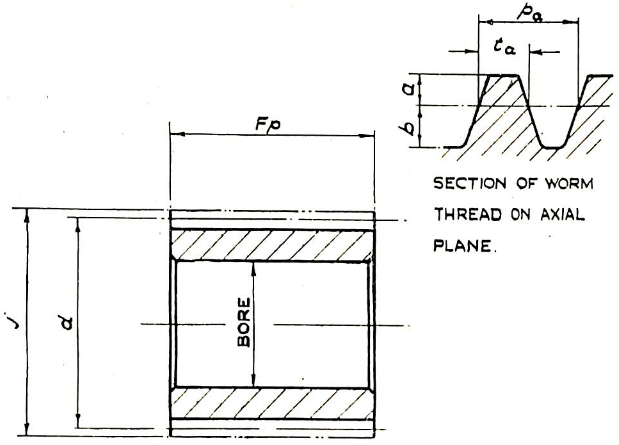
when $\tan \phi_c = \dfrac{p_a\, t}{d\, \pi}$

Note—ϕ_c is equal to the helix angle of worm and the spiral angle of the cutter.

$$L_c = \frac{D_c\, \pi}{\tan \phi_c}$$

$$B_o = b + y \tan 10^\circ$$

$$D_{oc} = D_c \cos \psi_{tc}$$



SECTION OF WORM THREAD ON AXIAL PLANE.

CUTTING PARTICULARS	
TYPE OF GEAR	WORM
HAND	R OR L
NUMBER OF THREADS	t
AXIAL PITCH	p_a
LEAD	L
AXIAL PRESS ANGLE	ψ_a
HELIX ANGLE	ϕ
ADDENDUM	a
DEDENDUM	b
NORMAL PITCH	p_n
NORMAL THICKNESS	t_n
CHORDAL ADDENDUM	a_c
CHORDAL THICKNESS	t_c
BACKLASH	—
RUNNING CENTRES	C
CUTTER NUMBER	HGS
TO GEAR WITH WORMWHEEL DRG. Nº —	

FIG. 20. PARTICULARS OF WORM TO BE CUT BY GEAR SHAPER CUTTER.

The cutter teeth are measured on the normal plane using a pair of gear tooth verniers and given t_{cn}.

$$\theta_n \text{ radians} = \frac{t_{cn}}{D_{cv}} \quad \text{when one radian} = 57{\cdot}296^\circ$$

and

$$D_{cv} = \frac{D_c}{\cos^2 \phi_c}$$

$$t_{cc} = \sin \theta_n \; D_{cv}$$

$$f = \frac{D_{cv}}{2}(1 - \cos \theta_n)$$

$$\text{then } B_{oc} = f + B_o$$

When checking the accuracy of the tooth profile a base disc corresponding to the base circle diameter D_{oc} is used in the same manner as that described for spur cutters, and the diameter of the pitch block required for grinding the tooth profiles of the cutter is obtained by using the same formula as that used for helical cutters.

Example.

The cutter particulars for a gear generating cutter to cut a worm 0·250 axial pitch single thread right-hand; 14° — 30′ axial pressure angle, 2·000 pitch diameter, with straight side flanks on the axial plane are given below, and shown in Fig. 21 in the form of a working drawing.

$a = 0{\cdot}0796$

$b = 0{\cdot}0921$

$p_a = 0{\cdot}250$

$t_a = 0{\cdot}124$

then Z_c from table $= 40$

$$\tan \phi_c = \frac{0{\cdot}250 \times 1}{2{\cdot}000 \times \pi} = 0{\cdot}03978$$

$$= 2^\circ - 17'$$

$$D_c = 3{\cdot}183 \text{ (same as wormwheel)}$$

$$\text{then } L_c = \frac{3{\cdot}183\ \pi}{0{\cdot}03978} = 253{\cdot}8$$

$$= 254 \text{ inches}$$

$$O_c = 3{\cdot}183 + 2\,(0{\cdot}0921 + 0{\cdot}05 \tan 10^\circ)$$

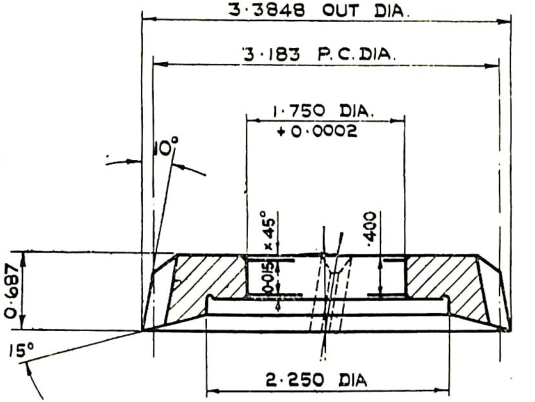
$$= 3{\cdot}3848$$

$$t_c = (0{\cdot}250 - 0{\cdot}124) + 2\,(0{\cdot}05 \tan 10^\circ \tan 15^\circ - 11')$$

$$= 0{\cdot}1308$$

$$B_o = 0{\cdot}921 + 0{\cdot}0050 \tan 10^\circ$$

$$= 0{\cdot}1009$$



MAT. H·S·S HARDEN & GRIND
LAP BORE.

FIG. 21. WORKING DRAWING OF GEAR SHAPER CUTTER FOR WORM.

CUTTER DATA	
TYRE OF CUTTER	WORM
PITCH	0·250
PRESSURE ANGLE	15°-11′
HAND OF CUTTER	RIGHT
NUMBER OF TEETH	40
ANGLE OF SPIRAL	2° - 17′
LEAD OF SPIRAL	254″
CUTTING DEPTH	0·1717
CHORDAL ADDENDUM	0·1022
CHORDAL THICKNESS	0·1306
BASE CIRCLE DIAMETER	3·0718
PITCH BLOCK DIAMETER	3·2593

ETCH: PITCH 14°-30′ PRESSURE ANGLE, CUTTING DEPTH, SPIRAL ANGLE, LEAD AND CUTTER Nº WGS. 19.

$D_{oc} = 3{\cdot}183 \cos 15^\circ - 11'$

$= 3{\cdot}0718$

$t_{cn} = 0{\cdot}1308 \cos 2^\circ - 17'$

$= 0{\cdot}1307$

$$D_{cv} = \frac{3{\cdot}183}{\cos^2 2^\circ - 17'}$$

$= 3{\cdot}188$

$$\theta_n \text{ radians} = \frac{0{\cdot}1307}{3{\cdot}188} = 0{\cdot}04099 \text{ radians}$$

$= 2^\circ - 20' - 54''$

$t_{cc} = 3{\cdot}188 \sin 2^\circ - 20' - 54''$

$= 0{\cdot}1306$

$$f = \frac{3{\cdot}188}{2}(1 - \cos 2^\circ - 20' - 54'')$$

$= 0{\cdot}0013$

$B_{oc} = 0{\cdot}0013 + 0{\cdot}1009$

$= 0{\cdot}1022$

To find the diameter of pitch block used to profile grind the cutter profiles shown in Fig. 21 when

$D_{oc} = 3{\cdot}0718$ and $L_c = 253{\cdot}8$ inches

$$\tan \beta_o = \frac{253{\cdot}8}{\pi \times 3{\cdot}0718} = 26{\cdot}300 = 87^\circ - 49''$$

$\psi_n = 20^\circ =$ angle of wheel head spindle

$$\cos \beta_G = \frac{\cos 87^\circ - 49'}{\cos 20^\circ} = 0{\cdot}04054 = 87^\circ - 41''$$

$$\therefore \tan \psi_{tG} = \frac{\tan 20^\circ}{\sin 87^\circ - 41} = 0{\cdot}36426 = 20^\circ - 01'$$

$D_G = 3{\cdot}0718 \sec 20^\circ - 01'$

$= 3{\cdot}2693$

$D_p = 3{\cdot}2693 - 0{\cdot}010$ when tape thickness $= 0{\cdot}010$

$= 3{\cdot}2593$

A.E.S.D. Printed Pamphlets and Other Publications in Stock.

An up-to-date list of A.E.S.D. pamphlets in stock is obtainable on application to the Editor, *The Draughtsman*, 96 St. George's Square, London, S.W.1.

A similar list is also published in *The Draughtsman* twice a year

Readers are asked to consult this list before ordering pamphlets published in previous sessions.

List of A.E.S.D. Data Sheets.

1. Safe Load on Machine-Cut Spur Gears.
2. Deflection of Shafts and Beams } Connected.
3. Deflection of Shafts and Beams (Instruction Sheet)
4. Steam Radiation Heating Chart.
5. Horse-Power of Leather Belts, etc.
6. Automatic Brakes (Axle Brakes) } Connected.
7. Automobile Brakes (Transmission Brakes)
8. Capacities of Bucket Elevators.
9. Valley Angle Chart for Hoppers and Chutes.
10. Shafts up to 5½-in. diameter, subjected to Twisting and Combined Bending and Twisting.
11. Shafts, 5¾ to 26 inch diameter, subjected to Twisting and Combined Bending and Twisting.
12. Ship Derrick Booms.
13. Spiral Springs (Diameter of Rd. or Sq. Wire).
14. Spiral Springs (Compression).
15. Automobile Clutches (Cone Clutches).
16. ,, ,, (Plane Clutches).
17. Coil Friction for Belts, etc.
18. Internal Expanding Brakes. Self-Balancing Brake Shoes (Force Diagram) } Connected.
19. Internal Expanding Brakes. Angular Proportions for Self-Balancing.
20. Referred Mean Pressure Cut-Off, etc.
21. Particulars for Balata Belt Drives.
22. ⅞" Square Duralumin Tubes as Struts.
23. 1" ,, ,, ,, ,,
24. ¾" Sq. Steel Tubes as Struts (30 ton yield).
25. ⅞" ,, ,, ,, (30 ton yield).
26. 1" ,, ,, ,, (30 ton yield).
27. ¾" ,, ,, ,, (40 ton yield).
28. ⅞" ,, ,, ,, (40 ton yield).
29. 1" ,, ,, ,, (40 ton yield).
30. Moments of Inertia of Built-up Sections (Tables) } Connected.
31. Moments of Inertia of Built-up Sections (Instructions and Examples)
32. Reinforced Concrete Slabs (Line Chart) } Connected.
33. Reinforced Concrete Slabs (Instructions and Examples)
34. Capacity and Speed Chart for Troughed Band Conveyors.
35. Screw Propeller Design (Sheet 1, Diameter Chart) } Connected.
36. ,, ,, ,, (Sheet 2, Pitch Chart)
37. ,, ,, ,, (Sheet 3, Notes and Examples)
38. Open Coil Conical Springs.
39. Close Coil Conical Springs.
40. Trajectory Described by Belt Conveyors (Revised 1949).
41. Metric Equivalents.
42. Useful Conversion Factors.
43. Torsion of Non-Circular Shafts.
44. Railway Vehicles on Curves.
45. Chart of R.S. Angle Purlins.
46. Coned Plate Developments.
47. Solution of Triangles (Sheet 1, Right Angles),
48. Solution of Triangles (Sheet 2, Oblique Angles).
49. Relation between Length, Linear Movement and Angular Movement of Lever (Diagram and Notes).
50. ,, ,, ,, ,, ,, ,, (Chart).
51. Helix Angle and Efficiency of Screws and Worms.
52. Approximate Radius of Gyration of Various Sections.

53. Helical Spring Graphs (Round Wire)
54. „ „ „ (Round Wire) } Connected.
55. „ „ „ (Square Wire)
56. Relative Value of Welds to Rivets.
57. Ratio of Length/depth of Girders for Stiffness.
58. Graphs for Strength of Rectangular Flat Plates of Uniform Thickness.
59. Graphs for Deflection of Rectangular Flat Plates of Uniform Thickness.
60. Moment of Resistance of Reinforced Concrete Beams.
61. Deflection of Leaf Spring.
62. Strength of Leaf Spring.
63. Chart showing Relationship of Various Hardness Tests.
64. Shaft Horse Power and Proportions of Worm Gear.
65. Ring with Uniform Internal Load (Tengential Strain) } Connected.
66. Ring with Uniform Internal Load (Tangential Stress)
67. Hub Pressed on to Steel Shaft. (Maximum Tangential Stress at Bore of Hub).
68. Hub Pressed on to Steel Shaft. (Radial Gripping Pressure between Hub and Shaft).
69. Rotating Disc (Steel) Tangential Strain } Connected.
70. „ „ „ Stress
71. Ring with Uniform External Load, Tangential Strain } Connected.
72. „ „ „ „ „ Stress
73. Viscosity Temperature Chart for Converting Commercial to Absolute Viscosities.
74. Journal Friction on Bearings. } Connected.
75. Ring Oil Bearings.
76. Shearing and Bearing Values for High Tensile Structural Steel Shop Rivets, in accordance with B.S.S. No. 548/1934.
77. Permissible Compressive Stresses for High Tensile Structural Steel, manufactured in accordance with B.S.S. 548/1934. } Connected.
78. Velocity of Flow in Pipes for a Given Delivery } Connected.
79. Delivery of Water in Pipes for a Given Head
80. (See No. 105).
81. Involute Toothed Gearing Chart.
82. Steam Pipe Design. Chart showing Flow of Steam through Pipes.
83. Variation of Suction Lift and Temperature for Centrifugal Pumps.
84. Nomograph for Uniformly Distributed Loads on British Standard Beams.
85. „ „ „ „ „
86. „ „ „ „ „ } Connected.
87. Notes on Beam Design and on Use of Data Sheets, Nos. 84-5-6.
88. „ „ „ „ „
89. Curve Relating Natural Frequency and Deflection
90. Vibration Transmissibility Curve for Elastic Suspension } Connected.
91. Instructions and Examples in the Use of Data Sheets, Nos. 89 and 90.
92. Pressure on Sides of Bunker.
93-4-5-6-7. Rolled Steel Sections.
98-99-100. Boiler Safety Valves.
101. Nomograph Chart for Working Stresses in Mild Steel Columns.
102. Pressure Required for Blanking and Piercing.
103. Punch and Die Clearances for Blanking and Piercing.
104. Nomograph for Valley Angles of Hoppers and Chutes.
105. Permissible Working Stresses in Mild Steel Struts to B.S. 449, 1948.

(Data Sheets are 3d to Members, 6d to others, post free.)

Orders for Pamphlets and Data Sheets to be sent to the Editor, *The Draughtsman*, cheques and orders being crossed "A.E.S.D."